
MACHINE DESIGN
September 1944
In This Issue:
Selecting Hydraulic Seals
Centralized Lubrication

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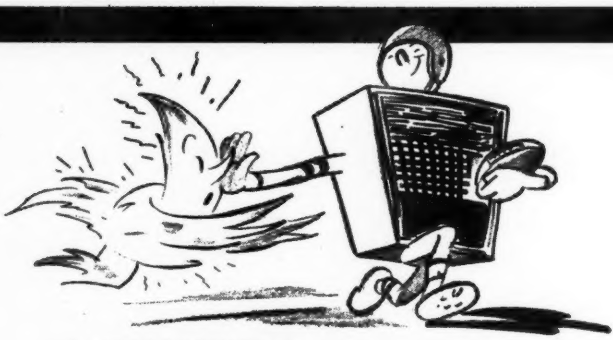
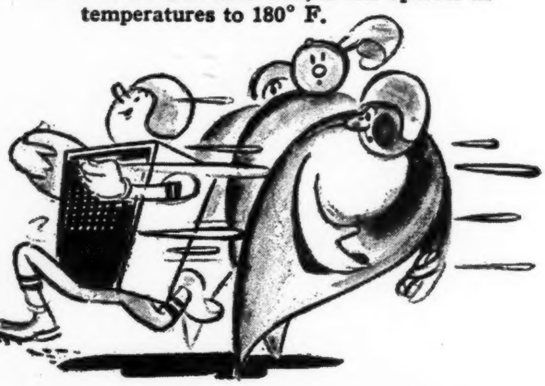
V-Belt Team!

ALLIS-CHALMERS recommendations for belt selections are unbiased . . . because Texrope's V-belt lineup is *complete.* And as inventor of the multiple V-belt drive, A-C offers sound engineering help . . . backed by the world's longest experience in giving industry money-saving drives.



1 HEAT-RESISTING SUPER-7 has ability to withstand high ambient built right into it. Standard belt, it can operate in temperatures to 180° F.

2 OIL-RESISTING SUPER-7 is right for drives where fairly bad oil conditions prevail. Neoprene-covered, it resists oil spray and grease.



3 OIL-PROOF SUPER-7, is solid Neoprene—a synthetic unaffected by most rubber attacking liquids. Recommended for *extreme* oil conditions.

4 STATIC-RESISTING SUPER-7 *conducts* static charge to machine for grounding charge. This safety belt is applied where explosion danger exists.

5 SUPER-7-STEEL employs endless steel cables to provide greater strength, reduce belt stretch. It is ideal for driving extra-heavy loads.

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MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

Contents

SEPTEMBER, 1944 **Volume 16—Number 9**

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MAIN OFFICE: The Penton Publishing Co., Penton Building, Cleveland 13.

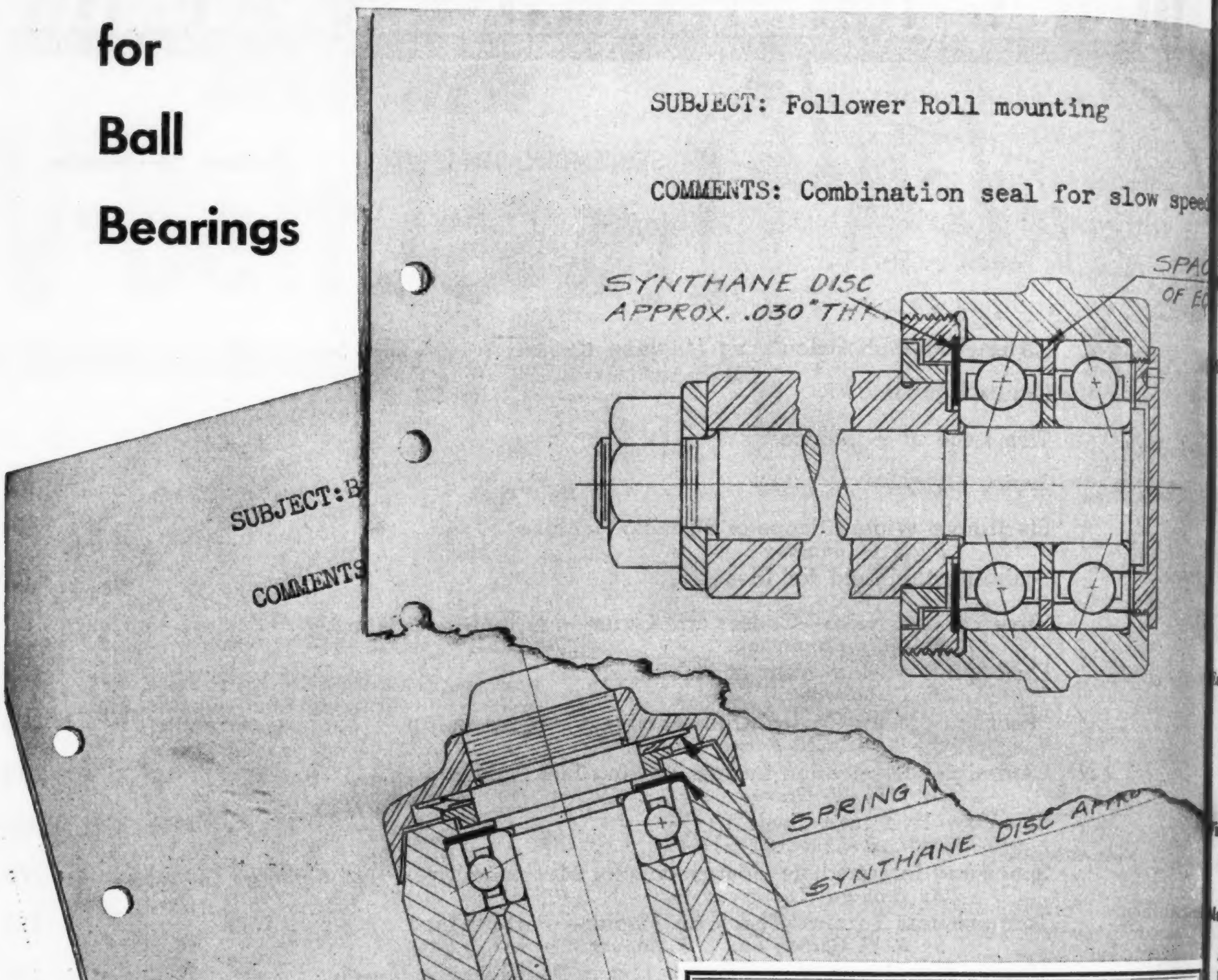
BRANCH OFFICES: *New York* 17, 16 East 43rd St., *Chicago* 11, 520 N. Michigan Ave.; *Pittsburgh* 19, Koppers Building; *Detroit* 2, 6560 Cass Ave.; *Washington* 4, National Press Building; *Los Angeles* 4, 130 North New Hampshire Ave.; *London S.W.* 1, 2 Caxton St., Westminster.

PUBLISHED BY The Penton Publishing Co. E. L. Shaner, Pres. and Treas.; G. O. Hays, Vice Pres. and Gen. Mgr.; R. C. Jaenke, Vice Pres.; F. G. Steinebach, Vice Pres. and Secy.; E. L. Werner, Asst. Treas. Published on seventh of month. Subscription in U.S. and possessions, Canada, Cuba, Mexico, Central and South America: Two years, $10; one year, $6. Single copies, 50 cents. Other countries: Two years, $14; one year, $8.

Acceptance under act of June 5, 1934, authorized July 20, 1943.

LOW COST LIFE ASSURANCE . . .

for Ball Bearings



THESE MOUNTINGS

operate at slow speeds and are subjected to considerable cutting oil or compound and abrasive matter.

Upper mount combines a labyrinth closure with a synthane sealing disc which presses lightly against the smoothly ground bearing face. Result is excellent protection against either wet or dry contaminants, without objectionable friction at speeds involved.

Lower mount swings through a wide arc and is thoroughly flooded with cutting oil. Speed of rotation is very slow. Under these conditions an extra sliding seal is used, consisting of a bronze ring held in light contact by a stock finger spring.

Seals such as these are relatively simple, occupy little space and are long-lasting. They also afford very good bearing protection and are efficient retainers for either oil or grease.

● Ball bearings will give long life and fine accuracy — but only when they are enclosed by methods which assure adequate protection from dirt or other injurious matter.

At slow speeds a positive seal is frequently essential. Here are two that are not expensive and are adaptable to many applications.

If you use or intend to use ball bearings, New Departure engineers are always ready to help you get the most out of them.

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Itemized Index

Classified for Convenience when Studying Specific Design Problems

MACHINE DESIGN is indexed in Industrial Arts Index and Engineering Index Service, both available in libraries generally.

This issue at a glance

Efficiency of Hydraulic Systems . . . .

. . . . can be hampered greatly by leakage. To keep hydraulic fluid—the lifeblood of such systems—within its confines, requires effective seals. The timely article beginning on Page 119 gives you the latest information on seal design.

Centralized Lubrication Is a "Must" . . . .

. . . . on some machines and a luxury on others. Where it fills the bill there are no fitting substitutes. What are the factors influencing type and quantity of lubricant, and type of metering valve employed? See Page 113.

Many Office Machines Have Seen Few Basic Changes . . . .

. . . . during the last decade. Improvements have been more in the nature of gradual refinements, utilizing the latest in styling, materials and processes. A machine in which such refinements—plus radical new developments—are incorporated, is discussed in the article beginning Page 91.

On Long Range Flights of Military Aircraft . . . .

. . . . accurate torquemeters are an absolute requirement. They aid mightily in keeping tabs on engine efficiency. What are the leading types? How do they work? Perhaps one would make your new machine a more efficient unit! See Page 135.

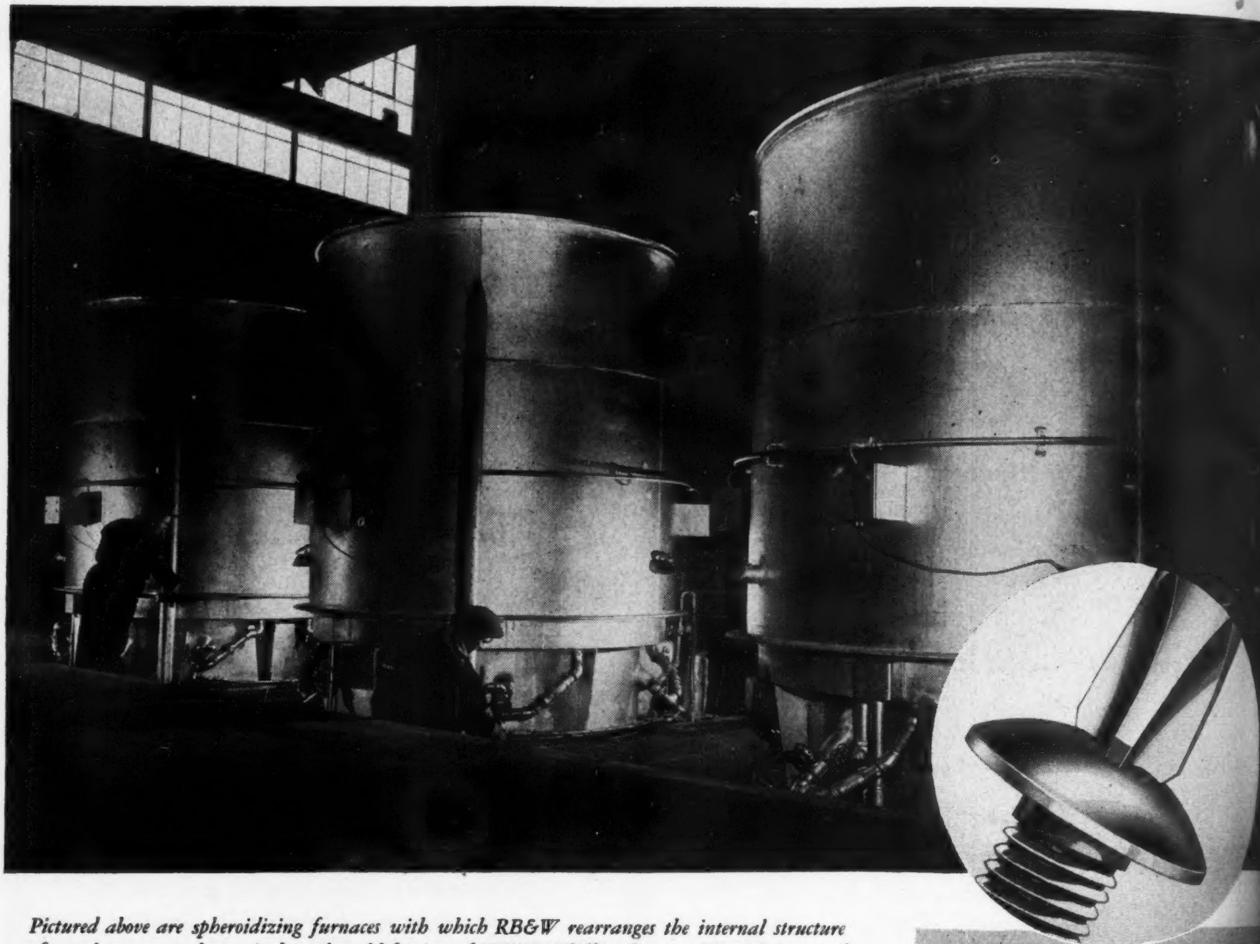
Wear of Metallic Surfaces . . . .

. . . . occurs in two principal ways. Projections and indentations of rough surfaces catch and rip; or galling, a form of incipient welding, occurs. The latter is caused by high surface pressure. If you're interested in an authentic analysis of the many factors contributing to wear of machine members, turn to Page 105.

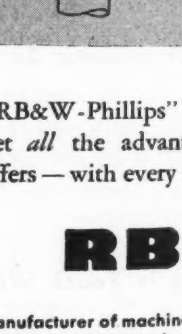
Keeping in Touch With the Field . . . .

. . . . is part of every machine designer's job. Latest trends and developments can exert profound influences on his work. Let "Outstanding Designs", one of the regular departments appearing each month in MACHINE DESIGN, help keep you posted. Turn to Page 140.

Making strong the things that make America strong



Pictured above are spheroidizing furnaces with which RB&W rearranges the internal structure of metal to a state best suited to the cold-forging of RB&W-Phillips Screws. RB&W is one of few screw manufacturers operating this type of equipment.



NEW SPEED WITH PHILLIPS SCREWS
from the oldest name
IN AUTOMATIC COLD-HEADING

Be sure you get *all* that Phillips offers — with every screw.

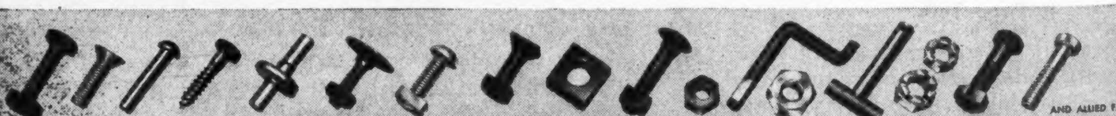
When you use "Phillips" instead of slotted, you're looking for faster driving — resulting from the quick, fumble-free start . . . use of faster driving methods . . . freedom from accidents.

To make sure of getting *maximum* driving speed — the full advantage of the Phillips head shape — specify "RB&W-Phillips", and benefit from the experience of the manufacturer *longest identified with automatic cold-forming.*

Additional savings in assembly time will come from RB&W's tough cold-drawn metal, strong cold-upset heads, clean cold-formed threads, accurate dimensions. The extra strength of "RB&W-Phillips" means that you'll get *all* the advantages that Phillips offers — with every screw.

RB&W

Manufacturer of machine screws . . . sheet metal screws . . . and stove bolts . . . with the Phillips Recessed Head.



EMPIRE BOLTS NUTS RIVETS AND ALLIED FASTENING PRODUCTS

RUSSELL, BURDSALL & WARD BOLT AND NUT COMPANY

Topics

EXPLOSIVE UNITS for emergency release of aircraft from their towed gliders are now being utilized by glider-towing planes in France and the China-Burma-India theater. Gliders can normally disengage the tow cable at the proper time for landing. Occasionally, however, an emergency may arise which requires the aircraft to disengage itself from the glider in a hurry. The explosive unit is located outside the plane and is operated by an electrical detonator within the plane.

FLAME-SPRAYED synthetic rubber on steel propeller shafts appears to reduce the menace of destructive electrolytic action, a problem of major concern to sub chasers and mine sweepers. Forming a firm bond, the rubber has high abrasion resistance.

PLASTIC FOAM, a new type of plastic developed by General Electric, is self-raising and self-curing. It is lighter than conventional insulation materials and lower in heat conductivity.

SYNTHETIC FLIGHT TRAINERS embodying the Dehmel patents are expected to prove an important factor, when fully developed, in permitting preflight analysis of new types of aircraft while they are still in experimental stages. The principles of "electromathics" as used in the trainer for simultaneous and continuous solution of complex problems are also applicable to other commercial products.

NEW ROCKETS much larger than the earlier "bazooka" projectile are being discharged from barges and other landing craft which mount strange multiple-barreled mechanisms to discharge several of the big rockets simultaneously. The rocket motor itself is composed of a long fuel chamber, a "discharge venturi" jet, directional guiding vanes and electrically-fired percussion contacts. Unlike the conventional heavy duty shells the velocity of the rocket accelerates while the projectile is traveling toward its objective.

THE 100,000 TURBOSUPERCHARGERS produced by General Electric have given Uncle Sam's fighting planes the means of recovering at high altitude more than 66 times the power generated at Boulder Dam, or almost twice the output of the entire nation's steam power plants combined.

ENOUGH POWER is packed in twenty Superfortresses, having four 2200-horsepower engines, to drive a 45,000-ton battleship of the Iowa class at a speed of 33 knots.

AUTOMATIC PILOTS have been adapted for use on Army gliders. By use of the device a glider can be set to follow the tow plane at a given altitude and on a true course, allowing gliders to be flown even in bad weather.

BASIC ENGINEERING DATA on castings, compiled by the S.A.E. War Engineering Board promises to aid in breaking the bottleneck to increased production of cast parts needed by the armed forces. Urging close cooperation from design to completion of castings, the recommendations suggest analysis of every detail and phase of manufacture. Proper inspection and selection of raw materials, segregation of scrap, correct operation of furnaces and other equipment, and quality control checks are also recommended.

HARDENABILITY DATA has been prepared by S.A.E. and A.I.S.I. in a booklet "Tentative Hardenability bands have been published. The steels, by hardenability is currently applicable to 37 standard fine-grain steels for which tentative hardenability bands have been published. The steels, identified by customary S.A.E. and A.I.S.I. numbers plus the code letter H, include 4100H from .30 to .50 carbon, 4340H, 4620H and the 8600H and 8700H series from .20 to .50 carbon.

Electronics Widens Scope of Business Machine

By S. G. Langley
Chief Engineer, Ediphone Div.
Thomas A. Edison Inc.

MANY of the most widely used machines of today are the result of evolution rather than radical new design. Their ancestries can be traced through a succession of steadily improving models each of which takes advantage of new construction methods, new styling and new technical developments in their own and related fields, while retaining the basic features characteristic of the machine. An outstanding example is the dictating machine family, the newest member of

Fig. 1—Electronic dictating machine includes Quadraphone (microphone and speaker) and recorder-reproducer cabinet

which, *Fig.* 1, is the subject of the present article.

Though often considered as a special recording device the dictating machine is, in reality, the most widely distributed recording equipment in daily use. The first phonograph was a recording machine developed for business purposes, *Fig.* 2. Since the development of this pioneer model these machines have been steadily improved, culminating in the recently introduced Edison electronic Voicewriter.

When it was decided to augment the services performed by the acoustic (voice-powered) equipment by designing an electrical machine involving amplification, experience gained through the manufacture of special-

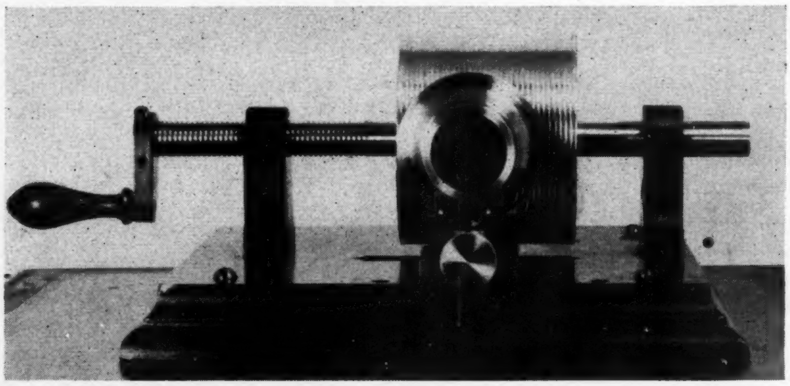


Fig. 2—Left—Original Edison tin-foil phonograph was the starting point of the sound-recording industry

Fig. 3 — Below — Side elevation shows mechanism which rolls carriage forward and opens letter-tray door

purpose electrical recording equipment (Telediphone) over a period of some twenty years was added to that obtained in the dictating machine field for sixty-seven years, with the result shown in *Fig.* 1. Development of this new machine may be roughly divided into three parts: Cabinet; amplifier and recording mechanism; and controls.

Cabinet design is based on that which was employed to provide complete enclosure of the 1933 model dictating machine. Mechanism which controls the mandrel and the recording and reproducing arm is mounted in a steel enclosure resting on the carriage shown in *Fig.* 3, which has rollers allowing the mechanism to be moved easily into operative and inoperative position by means of the hook visible to the left of the index slip in *Fig.* 1. As the carriage is rolled forward, the lever and link, *Fig.* 3, are put into operation by a pin which engages a slot, opening the letter-tray door pivoted on a pin at the side of the letter-tray support.

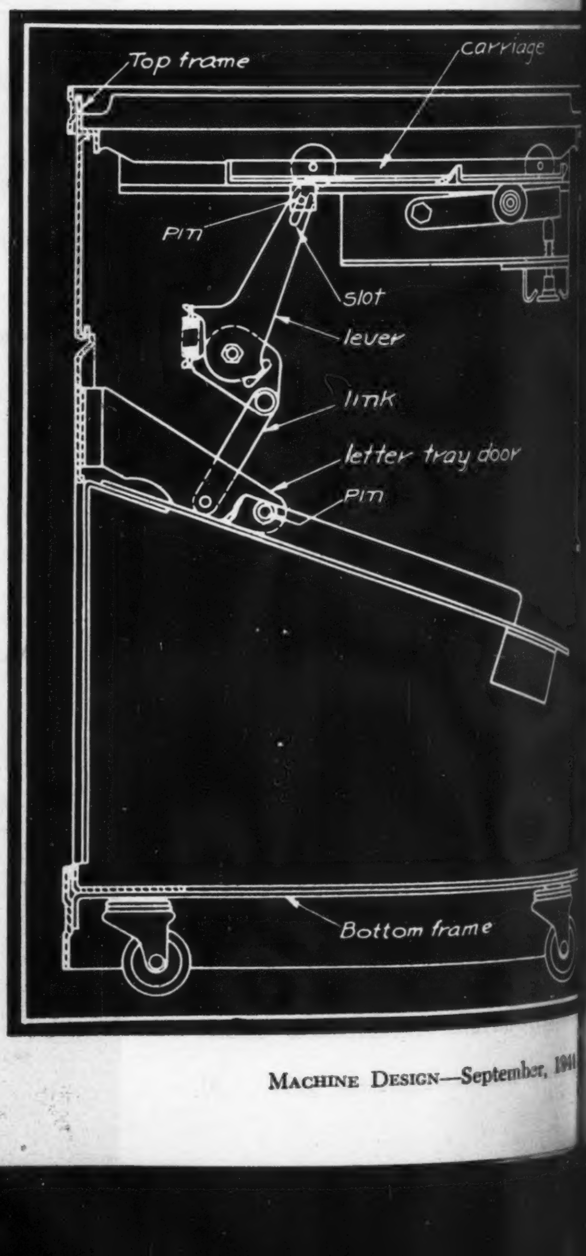
Slide Mechanism Performs Several Functions

Movement of the mechanism on its rollers permits the operating portions, including the control lever, the index slip and the length-and-correction punches to be brought fully forward and still keep the machine depth reasonable. Further, it permits a simple pivoted cover. The slide mechanism, *Fig.* 3, performs several functions. It serves as a support for the machine mechanism, it operates the master switch shutting off all power when moved back, and it opens the correspondence compartment door.

In *Fig.* 4 is shown the door (carrying correspondence folders and cylinders) in maximum open position, pivoted on a pin at the lower end and held from opening further by a stop. The same view shows the amplifier drawn outward on its rack as it would be when in need of servicing. The top cover is shown raised, the circular cutout being closed by a shutter when the cover is down.

Steel is used throughout in the construction of the cabinet. The body, shown in *Fig.* 5, is made of number 20 B & S gage auto body stock. After notching, piercing and forming it is completed by spot welding the three front members, which are held accurately in position by a fixture. Holes are jig drilled in the inwardly projecting flanges at top and bottom to which are bolted the upper frame containing the mechanism carriage and lower frame or base, which carries the casters.

Top frame, *Fig.* 3, is made from a strip of No. 20 B & S



gage steel, pierced and notched while flat, then flanged and formed rectangular to coincide with the top of cabinet body. The ends are held together by a plate spot welded to them on the inside while held in position by a fixture. To the outer upturned surface of the frame is spot welded a strip of round edge steel, No. 19 B & S gage, rolled channel-shape which, when assembled to the cabinet, conceals the joint of cover frame and cabinet and also provides a decorative effect. The depressed central groove in this strip is used for spot welding to the frame, then lacquered, thus obliterating the spot welds. The two raised portions of the strip are brightly polished.

Bottom frame or base, *Fig.* 3, is made of No. 16 B & S gage steel. It is sheared to rectangular shape, then the necessary holes are pierced and the corners notched out to allow forming four sides or flanges, resulting in the

Fig. 4—Space and heat dissipation for amplifier assembly, shown extended for servicing, was a major design problem

shape of an inverted pan having open corners. Following the construction used in the top frame, a round-edged channel-shaped strip of No. 16 B & S gage steel is formed into rectangular shape and then spot welded to the downwardly projecting flanges of the base. When assembled to the cabinet, the strip conceals the joint between the cabinet body and the base and is wide enough to conceal the casters.

Cover is made of three pieces of No. 20 B & S gage steel, consisting of right and left sides and a curved center panel accurately assembled into a fixture and held for spot welding. The cover frame, in which it is pivoted, is made of four pieces of the same material, formed to shape, fixture-held and spot welded.

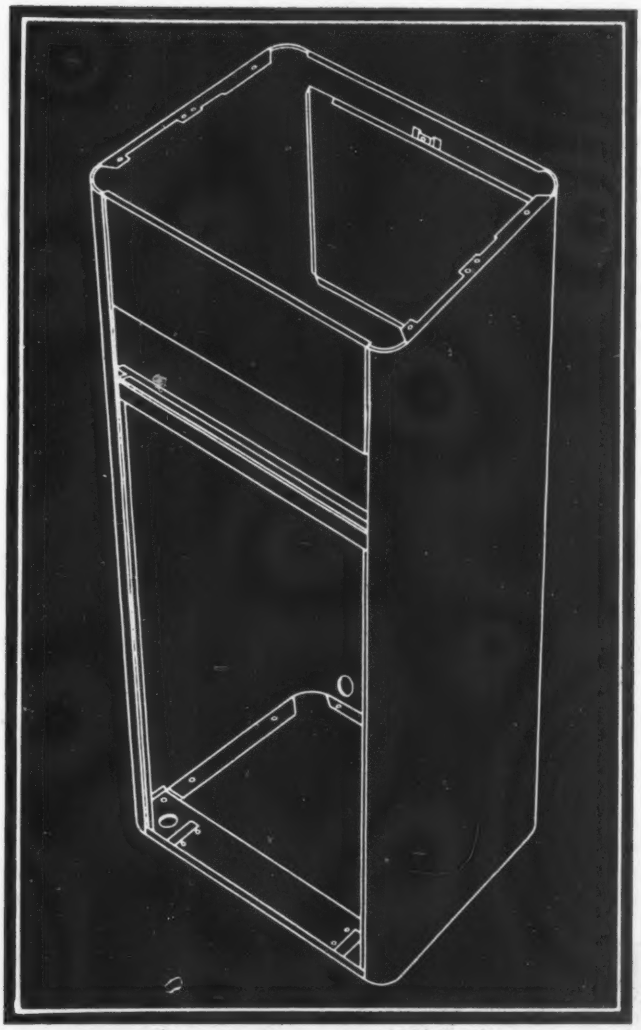


Fig. 5—Cabinet body is sheet steel formed and spot welded in fixtures, insuring accuracy plus or minus .015-inch. Holes for bolted attachments are jig drilled

Spot-welding fixtures used for producing these assemblies received careful consideration in their design, with the result that the parts consistently come within the tolerance of $\pm$.015-inch.

A principal problem was to obtain space and heat dissipation in the cabinet for the amplifier and control equipment. A portion of the back of the cabinet was cut out and a sliding rack inserted, *Fig.* 4. The electronic tubes were placed in the back of the amplifier near the ventilating grille. The interstage and transformer shield cans were arranged to act as a backing and heat-directing shield, a plate in the cabinet directly above the amplifier completing the heat-isolating system. The cabinet cover was arranged to open only part way and a small shield added to provide room above the machine mechanism for the electrical recorder and some of its controls.

Amplifier, shown in *Fig.* 4, is rather unique for it is a high-gain (112 decibels) AC-DC, three-stage design on a 9-inch by 7-inch chassis. As business office equipment often is accidentally disconnected from the power line at night by janitor service, it was necessary to make its operation independent of the polarity of the line, with the machine cabinet grounded or floating. There is no connection between the cabinet and the power line. This amplifier is further distinguished by a fast-acting, wide-

range automatic volume control and by an input transformer. This patented transformer is triple magnetically shielded and is so well electrostatically isolated that 120 volts, 60 cycles, can be connected between its primary and secondary without disturbing its operation. This permits the input system to be completely isolated from the power line and to be grounded to the cabinet.

Combination recorder-reproducer, *Fig.* 1, employs a Rochelle salts crystal as its conversion element. The crystal is clamped between two aluminum plates, and has a recording sapphire cemented to its end. The sapphire is mounted in a light-weight molded Lucite holder. For reproducing, a sapphire ball stylus is mounted in a small pivoted holder which coacts with the recorder stylus holder through a spring. This system results in a stiff recorder and a compliant reproducer.

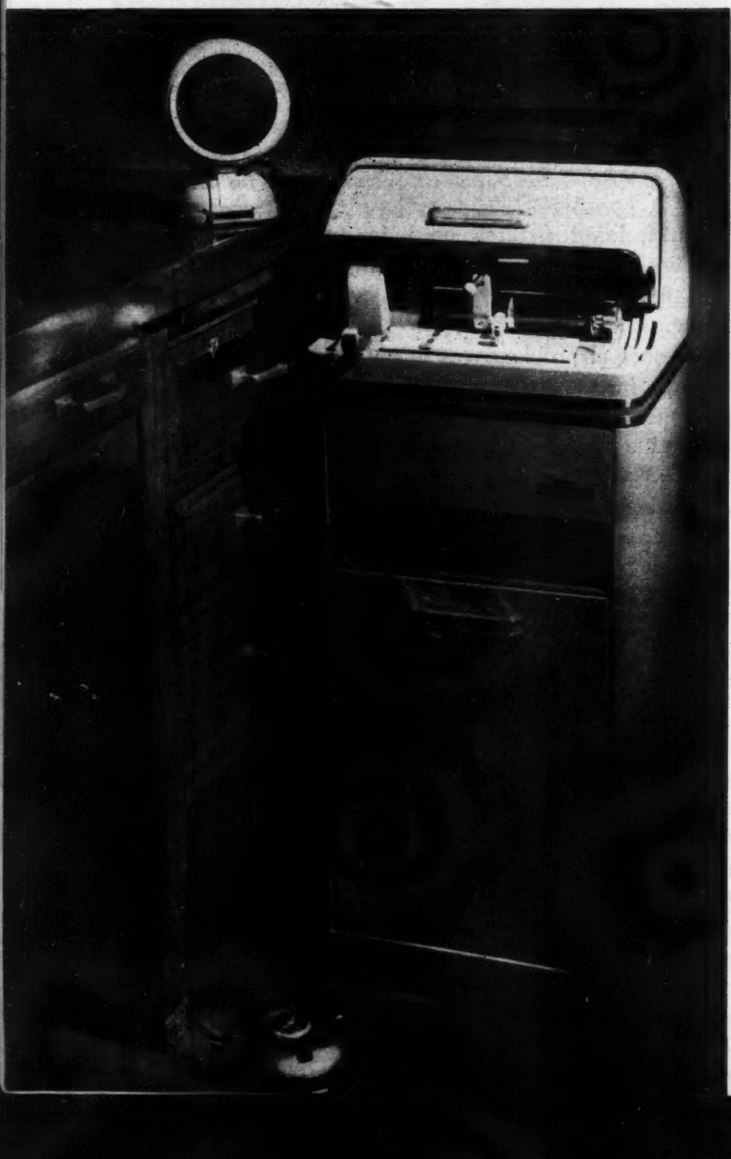
The wax cylinders employed are similar to the master wax disks used in making commercial phonograph recordings. Their surface noise is the lowest for any known material. This allows the recording to be made at extremely low levels, of the order of .00001-inch for the average, and still maintain a good "signal to noise" ratio. This may be compared to the .001-inch swing of commercial phonograph records.

To those not familiar with the field of recording, it might be of interest to make a few other comparisons. The recorder pressure on the record is approximately 3 grams (about 1/10-ounce). The reproducer pressure is only 10 grams (1/3-ounce), although the best home phonograph reproducer requires 1-ounce pressure.

Inasmuch as a system for use by business men must function every time and must require a minimum of controls, the control system has been made extremely simple and dependable. Essentially, it consists of a desk unit (Quadraphone), a floor switch, record-reproduce lever and an index.

Heart of the system is the Quadraphone. It contains a crystal microphone, a 5-inch permanent magnet speaker, a neon signal light and a service selector switch. The signal light is operated by the amplifier power supply and performs three functions: It shows when the amplifier is "hot"; a steady light indicates machine is conditioned for recording; and a flashing light indicates that the machine is in neutral or reproducing condition.

Fig. 6—Floor switch, used during dictation to start and stop the record, is bypassed when service selector switch in base of Quadraphone is in conference or external positions permitting continuous operation without attention



Controls Recording and Reproducing Circuits

Service selector switch in the base of the Quadraphone, *Fig.* 1, connects the amplifier input through a volume control to the microphone for dictation, (D), through a second volume control to the microphone for conversation recording, (C), and through a third volume control to an external circuit, (EXT), for recording speech originating at points remote from the machine such as radio or telephone. At the same time that the three operating conditions are set up, the motor is turned on. In addition, this switch connects the floor switch shown in *Fig.* 6 into the circuit for control during dictation and bypasses it in both conversation and external positions. This permits continuous operation on the record without manual attention.

Design of this unit presented some extremely interesting problems. The output and input circuits of the high-gain amplifier and the power line all come together and are confined in a very small space. Furthermore, the switching required a 3-pole, 4-position switch with complete shielding for 112 decibels between parts. In addition, one of the blades in one position had positively to break its circuit before making the next.

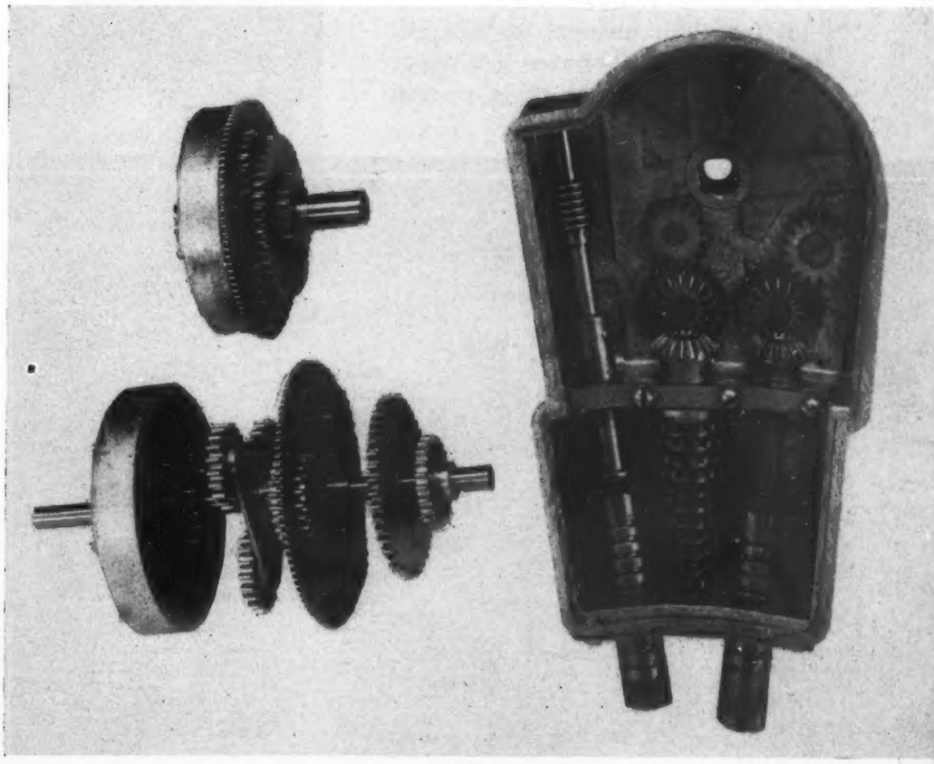
Die castings were used for the Quadraphone assembly to act as shields and mounting means as well as to obtain a pleasing and distinctive appearance. The floor switch operates a clutch to start and stop the cylinder through an electrically set, mechanically locked, electrically reset relay. The motor operates continuously—except when the Quadraphone switch is in "off" position—and clutch operation results in the cylinder reaching full recording speed in .05-second or less.

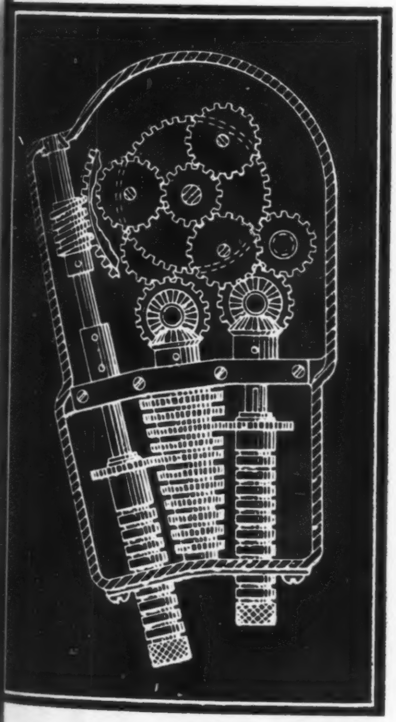
While retaining the features and conveniences of all previous models used for dictation, new facilities provided by electronics are made available, such as dictation at a distance, recording of conferences and interviews, recording of telephone conversations, all with immediately amplified playback.

From the foregoing brief discussion it will be appreciated that the successful design of this type of equipment involves the careful application and correlation of the principles of many fields of engineering.

Scanning the Field for Ideas

GEAR ratios having 121 variations are obtained with the positive-drive, variable-speed ansmission illustrated at right and elow. Designed by Ralph N. rodie Co. Inc. to calibrate liquid eters, the transmission utilizes a lanetary system with a changeear stack and two shiftable idlers. oth the shiftable idlers and the ear stack control the positions





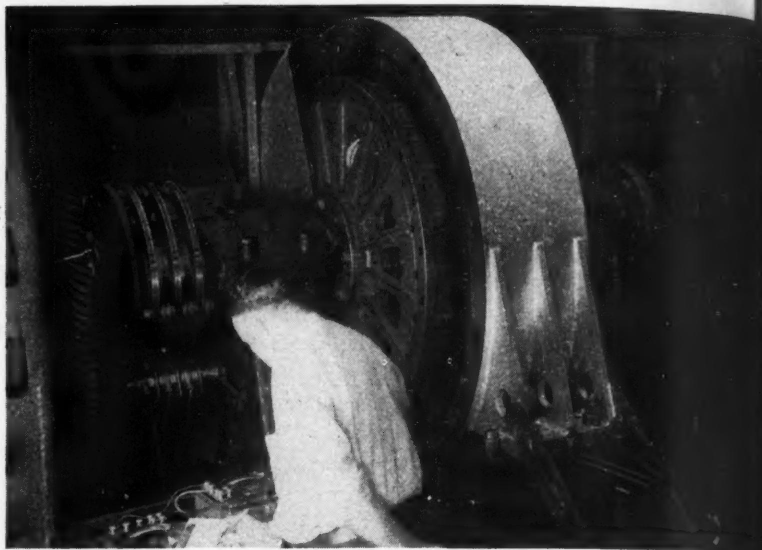
f the planetary gear elements give the variable ratios. The sysm illustrated was designed with a ominal input-output ratio of 1 to 4, e ratios varying between 1 to 9005 and 1 to 4.173. Increments r the coarse adjustment idler are 25, obtained by shifting the idler aving bevel gearing to an adjacent gear on the stack. Fine adjustment increments of .00225 result from shifting the idler having the worm gear. These fine increments are obtained for eleven positions leaving an increment of .0025 for the twelfth position, obtained by a shift of the coarse adjustment idler. Eleven such positions for the fine adjustment idler are available for each of the eleven shifts for the idler for coarse adjustment, making 121 ratios.

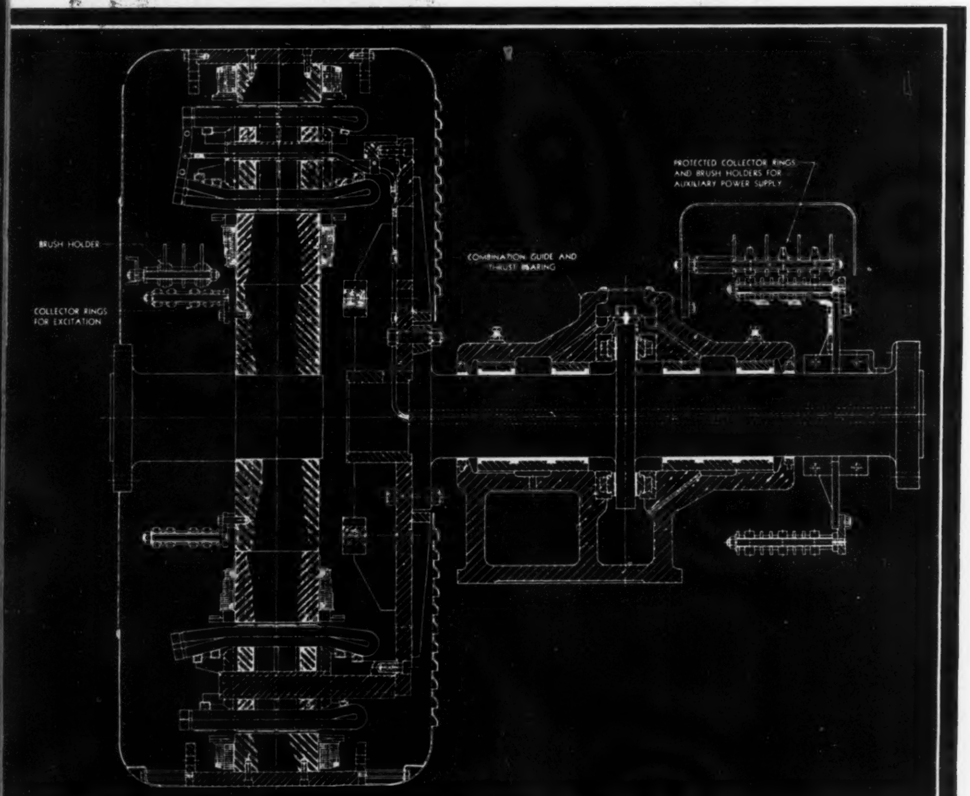
Electric speed reducer serves, in effect, the functions of three separate units—coupling, speed reducer and electric generator. Known as the Bowes drive, it comprises three main elements one of which is coupled to the prime mover, another bolted to the driven shaft and the third is a stationary element. Essentially a generator rotor, the engine element has poles selected for the desired speed ratio. The shaft element consists of a rotating spider which supports two sets of windings.

Together with the engine element the inner winding of the

spider forms a generator and transmits torque to the driven shaft due to the reaction resulting from generation of power. The outer winding of the spider, solidly connected to the generator winding, makes use of the field of the third or stationary element to operate as a motor, converting electrical energy into mechanical energy for turning the driven shaft. The stationary element supports a synchronous motor field and a motor-starting winding as shown in the drawing below.

If the shaft element is held stationary the unit behaves like an ordinary generator which is particu-

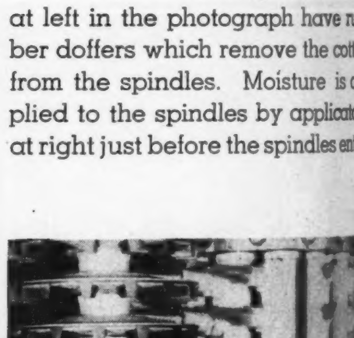


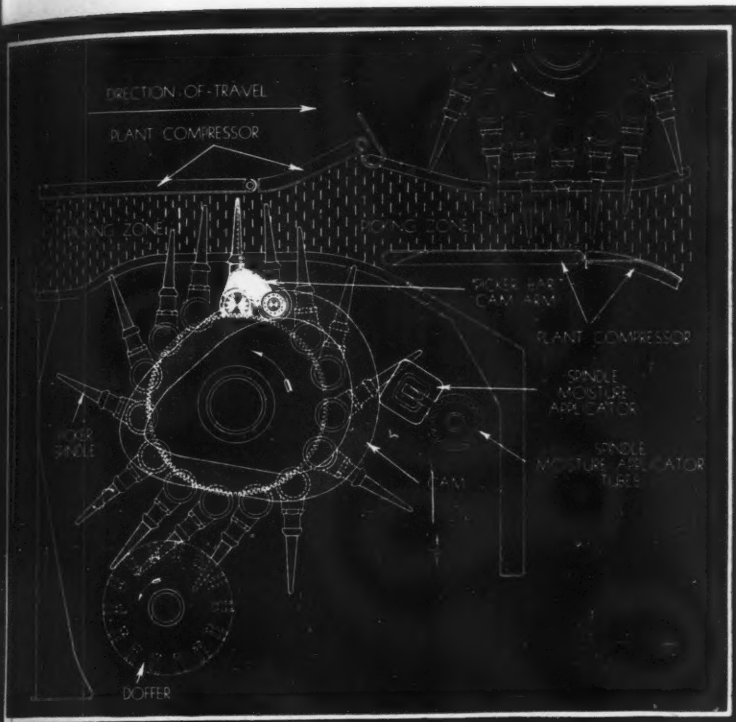


larly advantageous on ship propulsion systems where power is required in port. The drive illustrated above, built by the Elliott Co., is for three-to-one speed reduction on a 720 revolutions per minute, 900-horsepower diesel.

Approximately 15-kilowatt, three-phase power take-off is provided when the coupling is driving. With the shaft element locked, 400 kilowatts are available. A similar design is being developed for reversing duty.

Cam-actuated picker bars on the vertical drums shown in the photograph at right are utilized on the McCormick-Deering cotton picker. Fifteen bars are employed in a drum box and each have 20 spindles, making a total of 600 spindles per drum. Rotating at 2000 revolutions per minute the spindles are fitted with fine pointed barbs that have a high affinity for cotton lint. As the spindles pass through the picking area, cotton is wound around them and pulled out of the boll. Revolving disks shown at left in the photograph have rubber doffers which remove the cotton from the spindles. Moisture is applied to the spindles by applicators at right just before the spindles enter



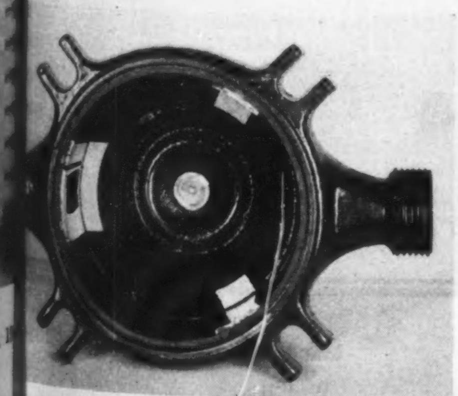


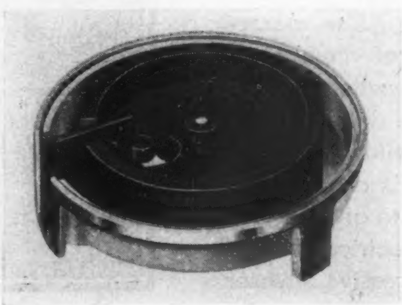
he picking area. Diagram of the echanism, above, shows the positions f the spindles as they pass the picking zone, doffer and moisture applicator. The cam action is so designed s to place the spindles at proper ngles for most efficient operation in ach position.

Automatic certification of hardess by stamping inspector's and heateat stamps on parts is performed by he tester illustrated at right. Considrable time is saved and the possiility of certification of a part of improper heat treatment is prevented. tamp mechanism utilizes a plunger nd an operating solenoid.

Mounted on the upright member is bracket supporting the testing mechnism and an electric motor. A block on the bracket carries bearing members which support a lever on a pin. This lever is urged upward by a strong spring to hold the follower against an eccentric driven by the electric motor which forces the indenter point of the hardness testing unit mounted on the other end of the lever into the material under test.

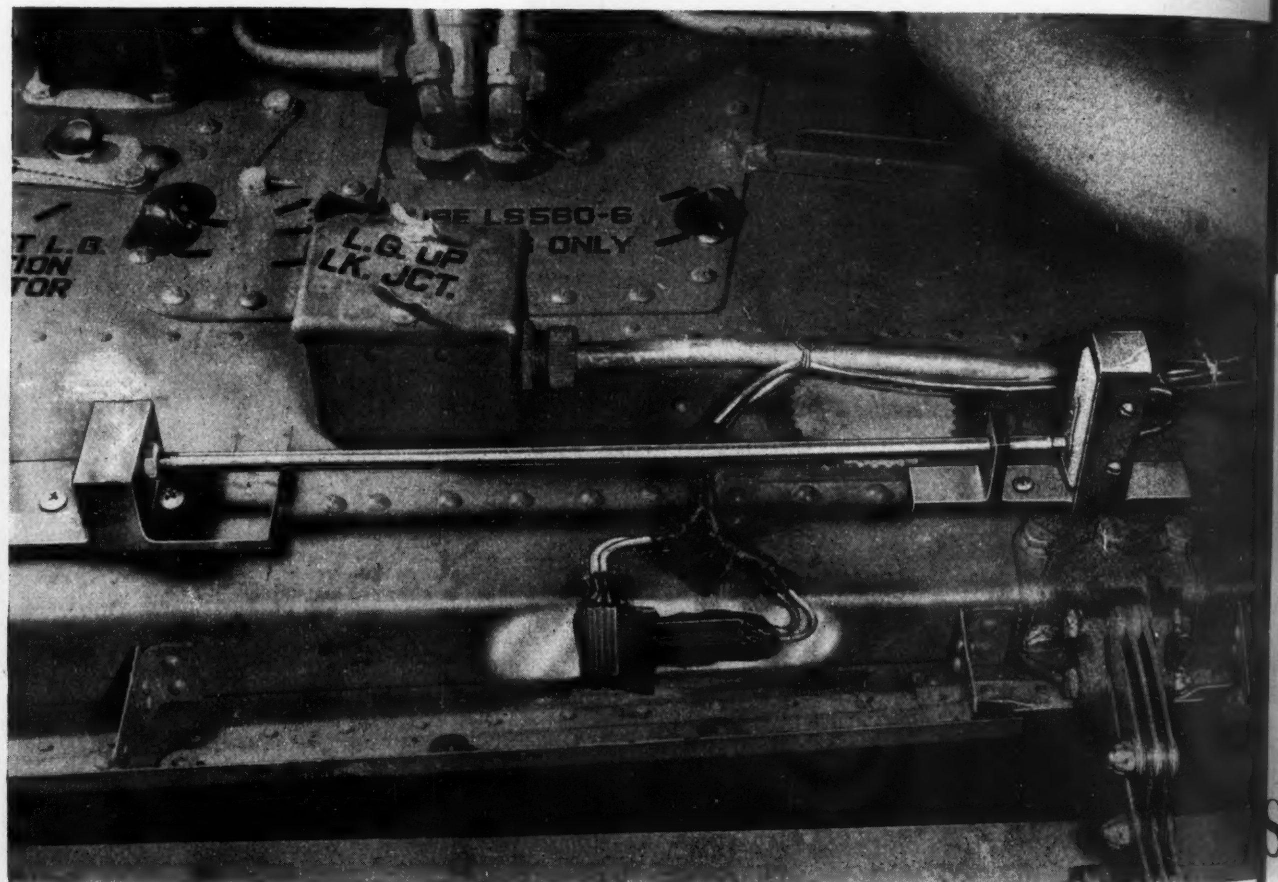
Contacts are secured in the indicator structure of a conventional testing unit and wired into an electrical circuit which includes a transformer connected to a source of power, a relay operated by the closing of the contacts which in turn closes other contacts, and a solenoid energized by the second set of contacts which actuates the stamping mechanism to mark the material. Operation is 43 times a minute in this mechanism, developed at Glenn L. Martin Co.





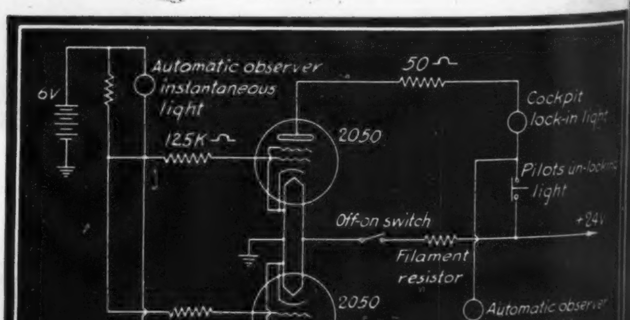
Poured-lead Inserts at points where the measuring chamber comes in contact with the cast iron case of the water meter shown at left provide jointing surfaces between these parts and prevent the formation of corrosion which otherwise would

make subsequent removal of the measuring chamber difficult. Exterior and interior surfaces of the case are protected by chemical treatment to inhibit formation of corrosion and by synthetic-resin enamel finish, baked at elevated temperature. With the return of use of bronze for case manufacture, the lead inserts and cast iron cases are no longer necessary. This design, however, built by the Pittsburgh Equitable Meter Co. shows how ingenuity may be employed to advantage to overcome limitations resulting from substitutions.



Stress warning for aircraft pilots, informing them immediately of dangerous structural conditions during flight, is accomplished by a simple but effective device developed by Lockheed and reported recently to the S.A.E. This unit, illustrated above, consists of five parts: Two clamps, a rod, a snap-type precision switch and an electrical circuit. The clamps fasten the unit to the structural member in question, holding the rod and switch in such a way as to transmit the relative deflection between the gage points to the switch. Calibration for desired stress warning is obtained by screwing the rod into the clamp to give desired gap for switch action. The electrical circuit, right, is designed to indicate excessive stresses even if they occur only for a short interval of time.

Two warning lights are used; one burns when the switch closes and goes out when it opens, the second locks-in and continues to burn after the switch is opened. Since normal relays require a few hundredths of a second to lock-in, vibratory stresses would not give an indication. Therefore thyratrons are employed which have a fast response. In the circuit employed, indications of four microseconds are sufficient for operating the indicating lamps. Pilot may unlatch the lock-in circuit if he wishes to know when stresses again become excessive during maneuvers.



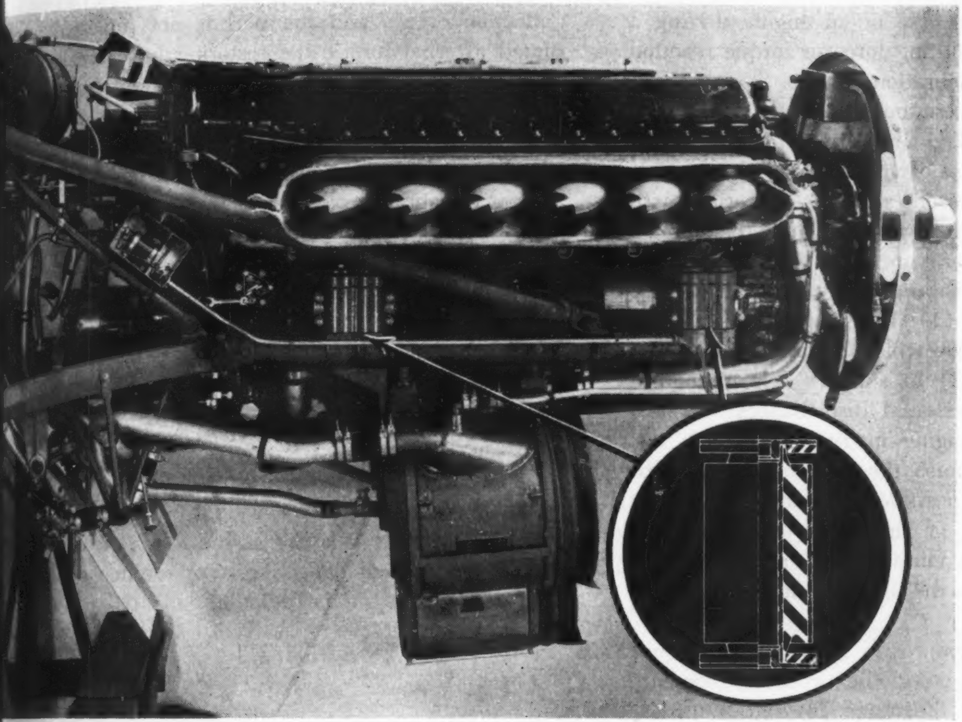


Fig. 13 — Left — Tube-form rubber mountings located at four support points of Allison engine on P-40 Warhawk isolate vibrations originating in the engine

Vibration and Noise—Causes and Cures

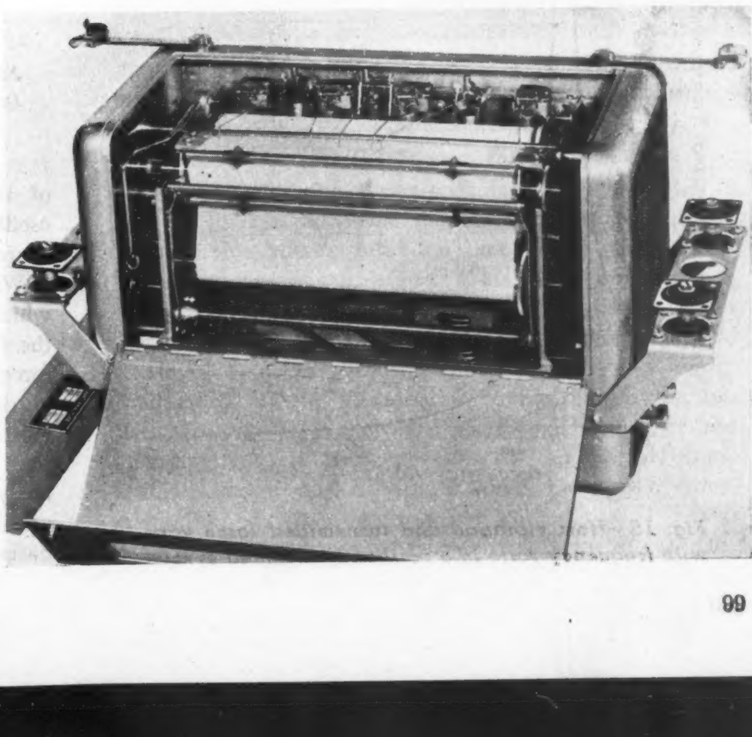
By Colin Carmichael

Part II—Isolation Mounting

Fig. 14—Below—Plate-form rubber mountings support the Friez flight analyzer, protecting it from disturbances due to plane vibration

SOURCES of vibration and noise in machines sometimes are so obvious at their magnitudes and frequencies can e accurately predicted. In other cases they re more elusive and can only be tracked down fter assiduous search with precision instruments such as were discussed in last month's rticle. In either case the proper procedure in ffecting a cure is first to eliminate or diminish s much as possible the source of the trouble, nd then—if vibration still exists—to isolate he source through the use of suitable materials r mountings.

Although much can be done toward eliminating vibration and noise at the origin by alancing rotating and reciprocating parts, ushioning impacts, etc., the residual unbalnce resulting from the necessary manufacturing tolerances still may be objectionable. urther, some types of machines deliver a eriodic force or torque externally which can-



not be balanced, as in the case of an engine driving a propeller, *Fig.* 13. The resulting force or torque reaction is an important source of vibration trouble.

It is therefore no exaggeration to say that there are few types of machines today whose operation could not be improved through the use of vibration-isolating mountings of some form. The present article is concerned with such mounting devices and how they should be applied to insure smooth, quiet operation.

Resilient mountings for vibration isolation are employed for one of two purposes: (1) To isolate from the surroundings those disturbances which originate within the mounted equipment, or (2) to support delicate equipment, protecting it from external disturbances. Typical of the first case is the engine in *Fig.* 13, which has resilient mountings to diminish the transmission of periodic torque reaction and possible unbalanced forces or moments to the structure of the plane. Protection of precision instruments from vibration in the surrounding structure is illustrated by the resilient mounting of a flight analyzer, *Fig.* 14.

The moving parts or pressures in machines which are in continuous operation set up periodic forces and moments of definite frequencies, as illustrated in *Fig.* 2 of last month's article. The resulting vibration is known as "sustained", "steady state" or "forced" vibration. It is due to the action of the periodic forces and moments on parts which are sufficiently elastic to respond.

Vibrations originating in mechanisms have a tendency to stimulate quivering of housings, panels, and other parts somewhat in the manner of a loudspeaker diaphragm. High-frequency vibrations and the higher harmonics of disturbances having lower frequencies therefore become audible as sound or noise. For this reason, isolation of vibration in a machine is a long step toward the suppression of noise.

Because machine structures possess both mass and elasticity, a deflection or deformation of any kind involves displacement of the mass and the creation of an elastic restoring force proportional to the deflection. When such

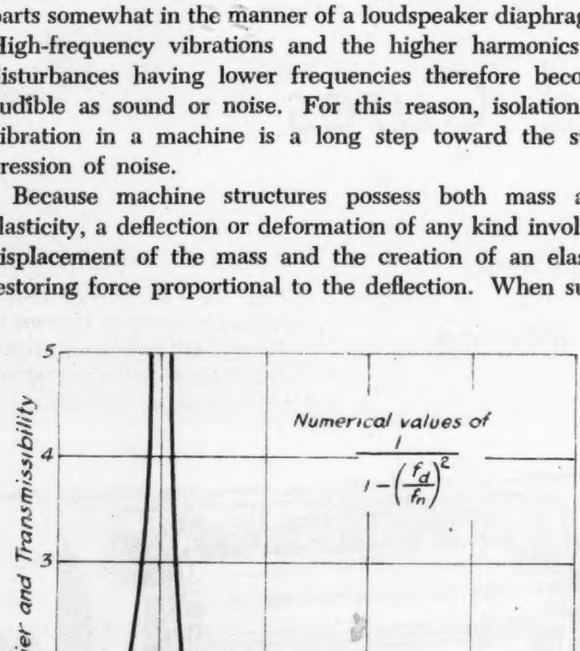


Fig. 15—How vibration and transmitted force vary with frequency ratio in a resiliently mounted system

deflection occurs and the part is not subjected to continued external forces, the restoring force accelerates the mass back toward the equilibrium or unstressed position with the result that when the mass reaches that position it is moving with some velocity and "overshoots the mark". The restoring force now acts in the opposite direction and a vibration cycle is set up which continues with diminishing amplitude until stopped by friction. The cycle repeats with a definite frequency, known as the natural frequency of free vibration, which is a function only of the mass and of the elasticity. The relationship for translational vibrations is

$$f_n=\frac{1}{2\pi}\sqrt{\frac{kg}{W}}=3.13\sqrt{\frac{k}{W}} \text{ cycles per second} \ldots\ldots$$

where W is the effective weight, pounds, of the part which moves and k is the stiffness or restoring force in pounds per inch of deflection. For rotational movement or oscillations a similar relation holds:

$$f_n=\frac{1}{2\pi}\sqrt{\frac{k_1 g}{Wr^2}}=3.13\sqrt{\frac{k_1}{Wr^2}} \text{ cycles per second} \ldots$$

where W is the weight as before, r is the radius of gyration

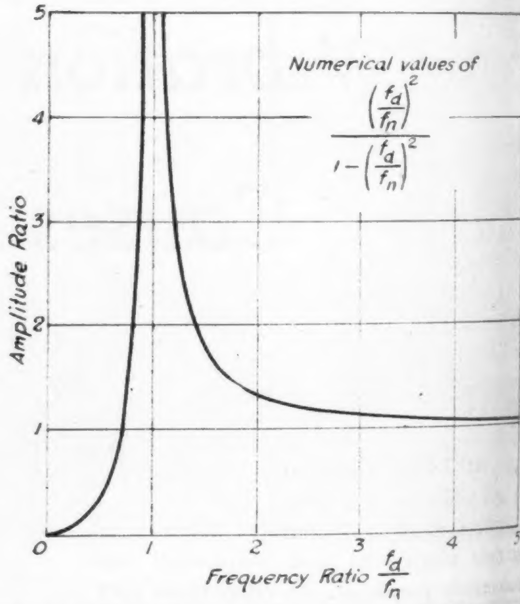
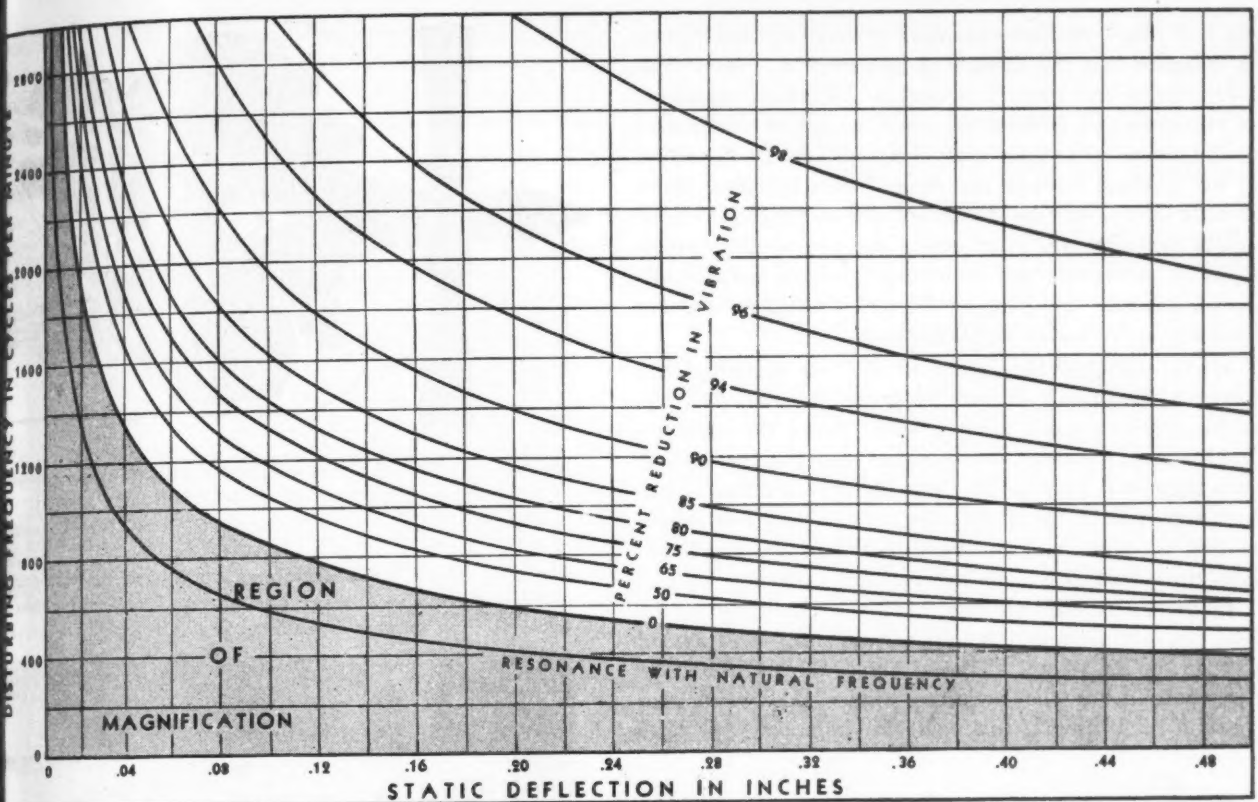


Fig. 16—Curve represents vibration amplitude as a function of frequency ratio for constant disturbing frequency and varying natural frequency

of the part with respect to the axis about which it oscillates, and k_1 is the rotational stiffness or restoring torque, inch-pounds per radian deflection.

When a periodic force acts continuously on the mass, which is the usual condition encountered in machines, the frequency of vibration will be that of the disturbing force rather than the natural frequency. But the ratio of disturbing frequency to natural frequency is of the most vital significance in determining the behavior of the part. While some mathematics is required for quantitative analysis of this forced vibration, it is not necessary to an understanding of the remarkable principle which is the basis of vibration isolation. This principle is simply



ig. 17—Chart may be used to select mountings for predetermined reduction in vibration

tilize the inertia of the part to oppose the disturbing rce. How this is accomplished may be understood from consideration of what happens when a periodic force applied to an elastically mounted machine, such as e engine in *Fig.* 13, for example.

When a slow up-and-down force is applied, the engine ill move up and down along with the force. Under ese conditions the disturbing force and the inertia of e engine act together, being opposed only by the restoring force of the mounting. If the disturbing frequency increased, the inertia force increases in proportion to the uare of the frequency, causing the vibration amplitude become greater. At the natural frequency of free bration the inertia force is great enough to balance the storing force of the mountings and there is, therefore, thing left to oppose the disturbing force. Continued eration at this speed—known as critical speed—results violent vibration which is limited only by friction or snubbing stops.

What happens if the disturbing frequency is increased ove the natural frequency? The inertia force is now eater than the restoring force which opposes it, and the ly way in which equilibrium can be attained is for e engine to start vibrating in opposite phase to the sturbing force. That is precisely what happens. Now e inertia force and the disturbing force oppose each her, the difference being balanced by the restoring rce at the mounting. As the disturbing frequency creases, the inertia force increases, and the difference hich the restoring force must balance becomes less. is means that the vibration amplitude becomes less, til at high speeds only a very small vibratory movement eates an inertia force equal to the disturbing force.

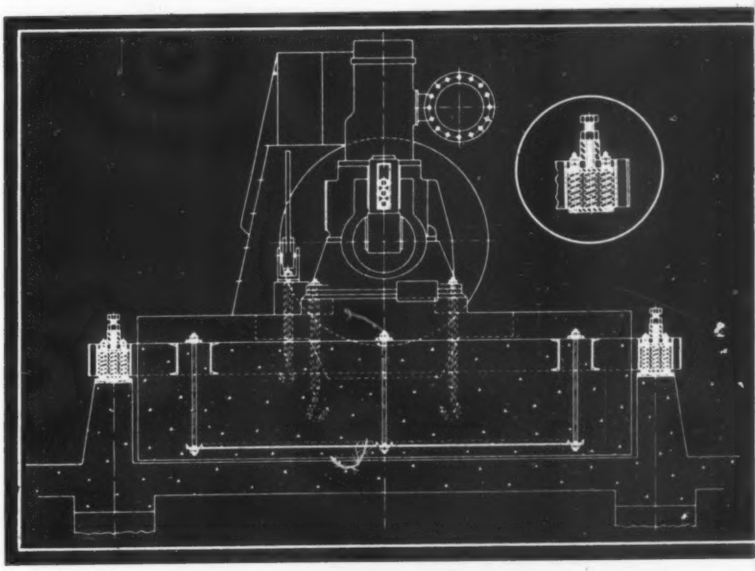


Fig. 18—With severe disturbing forces, vibration amplitude may be reduced by adding weight to the mounted equipment, as in this concrete base

How the vibration varies with frequency is shown in *Fig.* 15, which represents the amplitude of the motion as a fraction of the deflection that would result if the disturbing force were static. This ratio sometimes is called the "dynamic magnifier". Thus, if the maximum value of the disturbing force is P_0, the vibration amplitude would be P_0/k times the dynamic magnifier.

Inasmuch as the vibrating force transmitted through the mounting is proportional to the deflection of the mounting, *Fig.* 15 also represents the ratio of this transmitted force to the disturbing force, called the transmissibility.

Important conclusions may be drawn from *Fig.* 15. In

the first place, resilient mounting is beneficial only when so designed that the disturbing frequency is greater than 1.414 times the natural frequency. Further, inasmuch as vibration and transmitted force are greatly magnified at frequency ratios near unity (critical speed), operation of the machine through this range, when speeding up or slowing down, may be dangerous unless done quickly or unless suitable snubbing stops are provided. Certain types of mountings and mounting materials possess sufficient inherent damping or internal friction to prevent building up excessive amplitudes.

When studying the effect of various mountings of increased "softness" it must be borne in mind that the softer mounting has a lower k value, hence the vibration amplitude (P_0/k times the dynamic magnifier) is greater than suggested by *Fig.* 15. Actually the vibration amplitude with mountings of different stiffness varies as shown in *Fig.* 16, the amplitude being equal to the amplitude ratio times that in a "floating" system ($k=0$).

Inasmuch as the stiffness k of the mounting is the load per unit deflection, it may conveniently be expressed in terms of the weight of the machine, W, and the static deflection, d, of the mounting under this weight, or $k=W/d$. Substituting this value in Equation 1

$$f_n=\frac{3.13}{\sqrt{d}} \text{ cycles per second}$$

$$f_n=\frac{188}{\sqrt{d}} \text{ cycles per minute} \qquad (3)$$

This relation is represented in *Fig.* 17 by the lowest curve, marked "Resonance with natural frequency", corresponding to $f_d/f_n=1$ in *Fig.* 15. The next curve on *Fig.* 17, for 0 per cent reduction, corresponds to $f_d/f_n=1.414$, while the remaining curves are associated with frequency ratios greater than 1.414. This chart offers a convenient means for determining the mounting stiffness required for a predetermined amount of vibration reduction.

Though the foregoing discussion has been concerned with disturbances originating in the mounted equipment, the same principles apply to the isolation of equipment from external disturbances, and the curves in *Figs.* 15, 16 and 17 may be employed for the selection of mountings for either case. Two examples will serve to illustrate the use of these charts.

A single-cylinder diesel engine weighing 1200 pounds has a disturbing force of 300 pounds due to the inertia of the piston and connecting rod at its working speed of 1000 revolutions per minute. With rigid mounting, an alternating load of 300 pounds would be transmitted to the floor or deck.

Assuming that a reduction of 80 per cent is desired, leaving a transmitted force of only 60 pounds, reference to *Fig.* 17 shows that for 1000 cycles per minute the static deflection should be about .212-inch. A set of mountings which will deflect this amount under the 1200-pound weight of the engine would therefore be chosen. The natural frequency of the setup would be approximately 408 cycles per minute, (from *Fig.* 17 for static deflection .212-inch) giving a frequency ratio 1000/408= 2.45. The corresponding amplitude ratio, *Fig.* 16, is seen to be reasonably close to the ideal. The actual

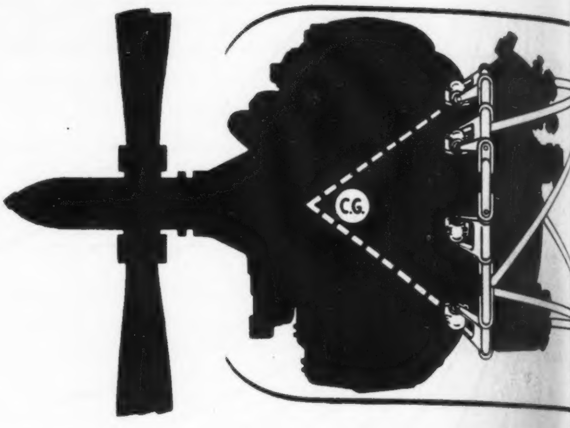
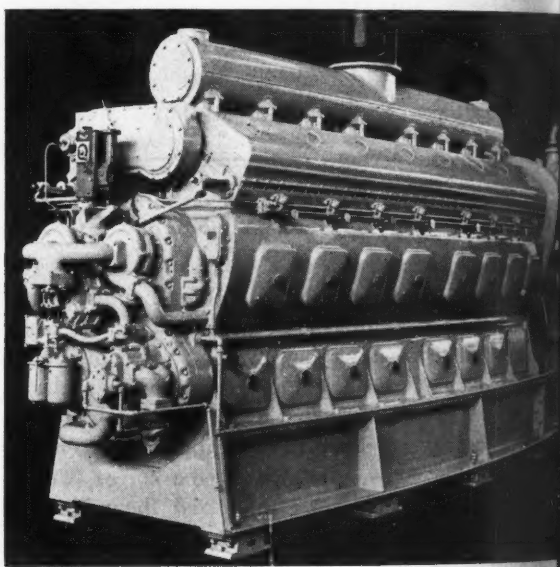


Fig. 19—Above—Engine mounting utilized in this installation provides virtual center of gravity support

Fig. 20—Below—Vibration isolators used on this marine engine incorporate steel springs and internal guides to withstand effects of rolling and pitching of the ship



vibration amplitude is readily figured by simple proportion. Since 1200 pounds deflects the mounting .212-inch, 60 pounds will deflect it .212 x 60/1200=.0106-inch, which is the amount of vibratory movement.

In the event that even this small vibration is too great, it may be reduced by adding weight to the engine, such as a concrete block forming part of the mounted structure, *Fig.* 18. Thus, assuming another 1200 pounds is added to the weight, for the same reduction in transmitted force the static deflection would be the same. However, to give the same static deflection the mountings would have to be twice as stiff as before because the mounted weight is doubled. The vibration amplitude would now be .212×60/2400=.0053-inch.

As an example of a mounted instrument, such as the flight analyzer, *Fig.* 14, it may be assumed that the point of support vibrates with an amplitude of ⅛-inch at a frequency of 1600 cycles per minute. If it is desired to reduce the movement of the instrument to not more than .005-inch, the reduction would be (.125-.005) /.125=.96 or 96 per cent. From *Fig.* 17, the static deflection should be .36-inch. Transmitted force would

$(.125+.005)/.36=.36$ times the weight of the instrument.

With a mounting which permits a certain amount of relative movement the question of "degrees of freedom" arises. A resiliently mounted part has six degrees of freedom—linear movement along three mutually perpendicular axes and rotary movement about these three axes.

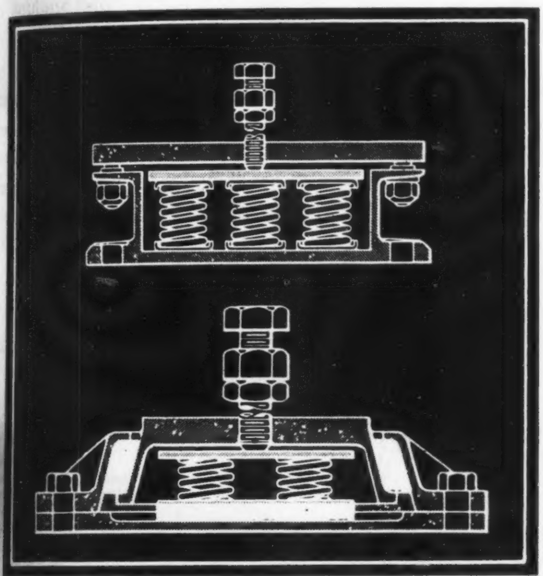


Fig. 21—Above—Mountings with and without dampers are two of several designs incorporating steel springs

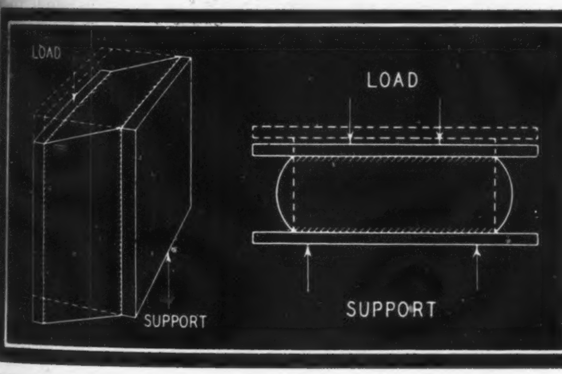
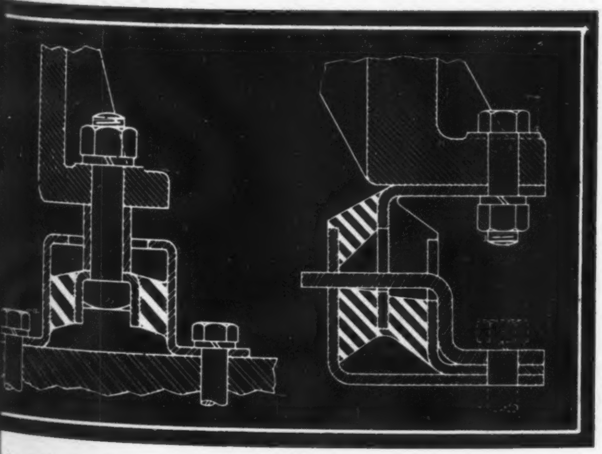


Fig. 22—Above—Basic methods of loading rubber are shear and compression. Rubber is practically incompressible, hence must be free to bulge as shown

Fig. 23—Below—Two types of shear mountings employing rubber bonded to channel or angle bars



Inasmuch as the mountings generally have different stiffnesses in each of the three directions, three different natural frequencies result from the three values of k in Equation 1. Further, the radius of gyration, r, is different about each of the three axes, as well as the rotational stiffness k_1 due to the mountings, giving three more natural frequencies from Equation 2. The first three "modes" of vibration are stimulated primarily by disturbing forces along the axes while the other three result from applied torques or moments.

When applying a mounting to isolate a vertical disturbance, for example, it is possible that the lateral stiffness of the mounting may be such that the mounted part has a natural frequency in the lateral direction close to the frequency of a relatively minor disturbing force. Severe lateral vibrations could therefore be set up unless care is exercised in selecting the mounting. The frequencies of all disturbing forces, no matter how minor, should therefore be known so that critical speeds can be avoided.

Because isolation depends on a mounting which gives a natural frequency less than the disturbing frequency, a mounting with lateral stiffness much greater than the vertical is a possible source of trouble. By installing mountings in series, as in *Fig.* 14, lateral stiffness can be kept low.

The presence of higher harmonics of the main disturbing frequency need cause no concern, inasmuch as their frequency ratios are greater than those of the fundamental and they are therefore more completely isolated.

Mounted equipment with a high center of gravity may show unfavorable performance due to instability, unless the mountings are elevated to approximately the level of the center of gravity. This is done in the instrument mounting shown in *Fig.* 14.

Avoiding Coupled Vibrations

If the line of action of the mountings passes through the center of gravity of the mounted equipment, disturbing forces stimulate translational motion only, while torques or moments cause only rotation about the corresponding axes. With any other arrangement of mountings there is the possibility of "coupled" vibrations occurring. This is the result of vibrational energy in one direction transferring to another direction, causing vibrations in a direction other than that of the disturbing force. Such transfers may be transient but they are not, on that account, any less troublesome.

A system of mounting giving virtual center of gravity support is widely used for powerplants in automobiles and aircraft. The chief disturbance here is the variation in torque reaction, and the mountings are applied so as to afford torsional resilience about a longitudinal axis. By choosing the axis about which the moment of inertia is least, virtual center of gravity suspension is attained.

Radial engines in aircraft are supported from one side only, leaving the center of gravity overhung. However, the directional stiffnesses of the mountings in a "Dynafocal" system, *Fig.* 19, provide center-of-gravity suspension characteristics, the individual mountings being inclined at a mathematically established angle so that they point toward a focal point just beyond the center of

gravity. Such a system is used on the B-29 bombers.

Unit type mountings employ either steel or rubber springs. The installations in *Figs.* 18 and 20 use steel springs, and details of two typical mountings are shown in *Fig.* 21. The unit employing steel springs alone has practically one hundred per cent resilience (no damping) and is therefore highly efficient as an isolator. It is designed for vertical thrust only. When slight lateral thrust must be absorbed and some damping is desired, the unit employing resilient material in addition to the spring, *Fig.* 21, often is preferred. For steel springs it is argued that the elasticity can be accurately controlled and will not change with time, while adjustments can be made after installation.

Rubber springs are built into mounting units in a variety of ways. Depending on the application, the material may be stressed in shear or compression, *Fig.* 22. With shear loading larger deflections are possible while with compression loading heavy loads can be carried and the mounting can be made relatively stiff.

Shear type mountings may be of tube or cylindrical form, as in *Fig.* 13, plate form, *Fig.* 14, or various forms of sandwiches. Two examples are shown in *Fig.* 23.

Compression type mountings, being relatively stiff, are suitable for reducing transmission of high-frequency vibra-

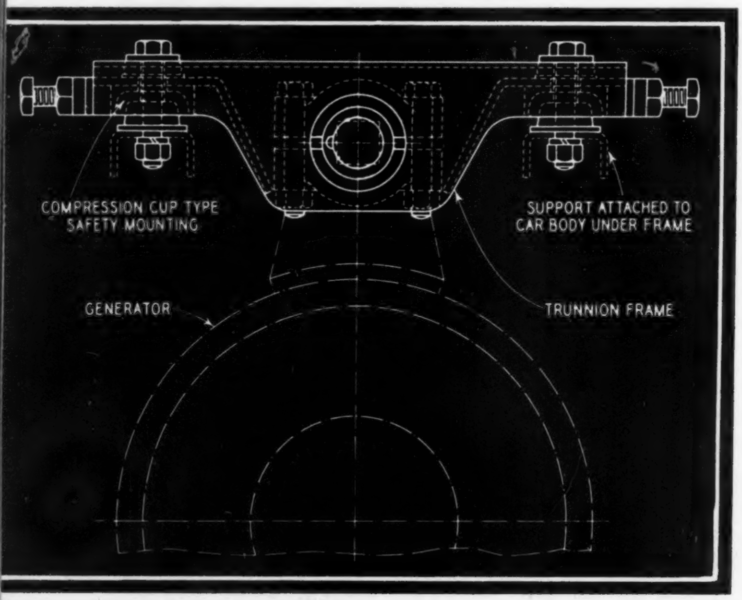


Fig. 24—Railroad car generator held by compression-cup rubber mountings so that noise is not transmitted to car

tions which might result in noise. An application is shown in *Fig.* 24, which illustrates a compression-cup mounting used on the generator under a railroad car, the purpose being to prevent sounds originating in the generator from being transmitted to the car body.

Certain forms of mountings can be applied either as shear or compression mountings. An example is the cylindrical type, an application of which is shown in *Fig.* 25, the mountings being employed to isolate police radio instruments from the shocks and vibrations of a motorcycle. In this case the weight of the equipment stresses the rubber in shear but transmitted forces may produce both shear and compression.

In shear mountings, the rubber usually is bonded to the metal by means of special adhesives. Mechanical adhesion, obtained by the pressure resulting from distortion of the rubber sleeve, is employed in some instances, notably in Silentbloc mountings. By controlling the amount of distortion, which is obtained by squeezing a separately molded rubber sleeve or biscuit into a metal sleeve of smaller diameter and then using a tapered mandrel to expand the inner diameter, the amount of "grip" can be made sufficient to resist slip.

Materials for rubber mountings may be of natural or synthetic rubber, compounded to give any desired characteristics such as elasticity, damping characteristics, heat and corrosion resistance, fatigue resistance, etc. In general the synthetics have somewhat greater damping or internal friction. This would be an advantage in preventing the

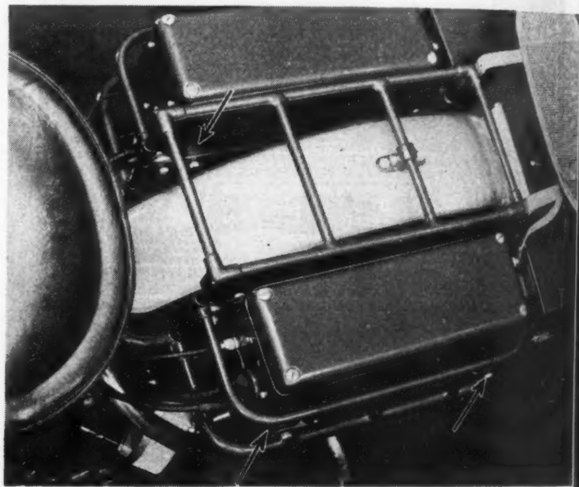


Fig. 25—Cylindrical type rubber mountings protect motorcycle police radio from vibrations and shocks

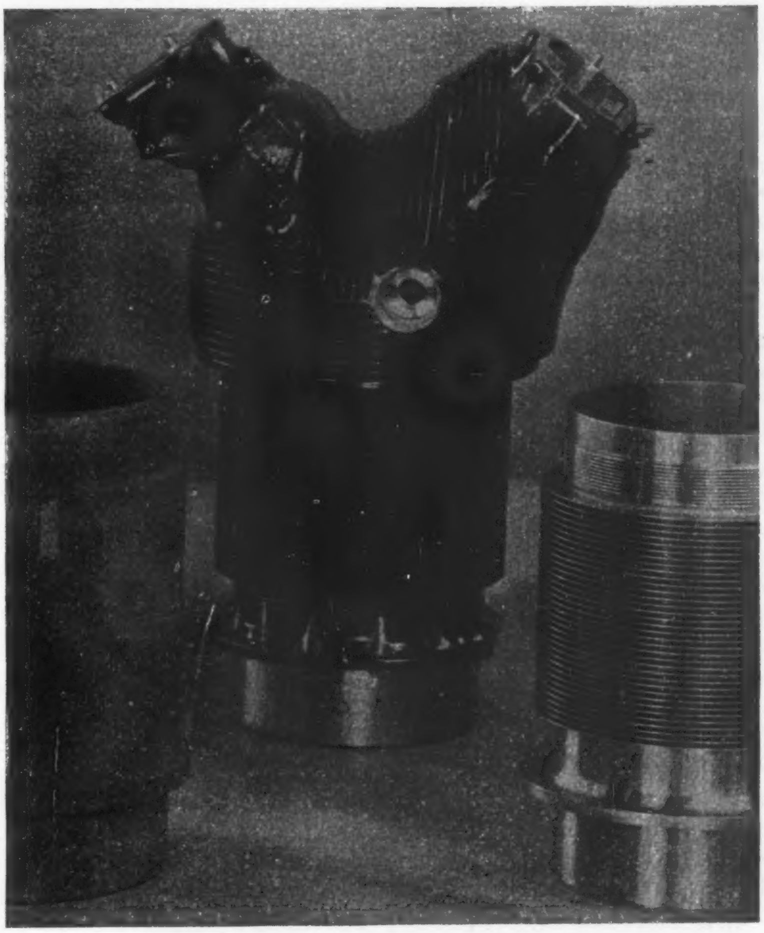
building up of high amplitudes while operating through the critical range, but of course continual operation in a range requiring energy absorption would tend to overheat the material.

With the increasing importance of vibration and noise control, it is becoming necessary for machine designers to provide for suitable mountings in their original designs. Too often the mountings have been added as an afterthought, with results which may fall short of the ideal. Much can be done, too, in isolating vibrations within a machine, rather than depending on resilient mounting for the entire machine. An outstanding example of this is the domestic refrigerator, the compressor of which is so effectively isolated from the cabinet itself that no other provision need be made. This principle might well be extended to many other types of machines.

In addition to unit type mountings such as have been discussed in this article, there are many materials which can be applied in the form of sheets, pads, slabs, etc., to isolate vibration and noise. These will be discussed in Part III of this series.

MACHINE DESIGN acknowledges with appreciation the cooperation of the following in the preparation of this article: General Tire & Rubber Co.; The B. F. Goodrich Co.; Harris Products Co.; The Korfund Co. Inc. (*Figs.* 18, 20 and 21); Lord Manufacturing Co. (*Figs.* 13, 14, 17 and 19); E. I. duPont de Nemours & Co.; United States Rubber Co. (*Figs.* 22, 23, 24 and 25).

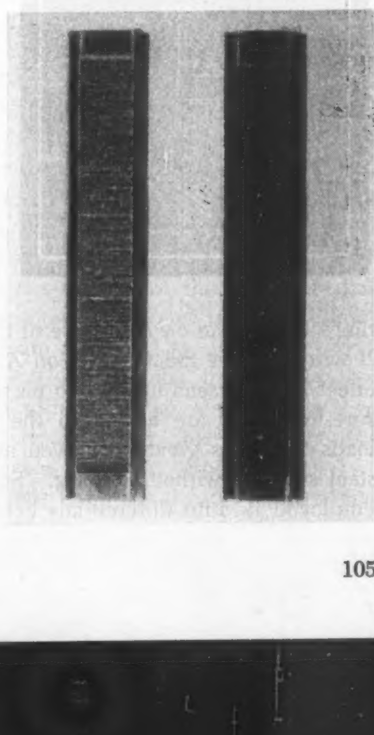
Factors Influencing WEAR in Machines



By D. Landau
Industrial Applications Engineer
The Nitralloy Corp.

Fig. 1 — Above — Aircraft engine cylinder barrels are subject to severe wear conditions, including abrasion, erosion and corrosion

Fig. 2—Below—Nonlubricated cast-iron specimen at left showed 260 times faster wear than lubricated specimen at right which carried 65 times as great a load



THERE are two principal ways in which wear occurs in metallic surfaces. The asperities of one of the surfaces either catch on the asperities of the other, or penetrate into the hollows of the mating surface; particles of either, or both, of the surfaces are torn off when the surfaces slide away from one another. In the second method of wear the mating metals come into very close—molecular—contact owing to the high surface pressure, and a sort of incipient welding (galling) occurs. When such surfaces are separated they must be torn apart, with the particles of one adhering to the other, resulting in wear.

Many attempts have been made to find some relation of wear to other physical constants of the metals, but wear apparently is not controlled by any one property. When similar materials are compared, the chief criterion for wear is hardness. In some instances cold working, melting temperature, and even thermal conductivity are the factors which index wear.

Surface Melting: In some cases it is possible to carry such high specific loads, together with a rate of surface rubbing speed, as will produce heating of sufficient magnitude to cause surface fusion or melting. A phonograph needle rubbing against a wax record may reach a temperature as high as 2000 degrees Fahr. (1)* at the immediate contact point where the needle touches the disk. Of course, the depth of this temperature is not more than a few molecules in thickness. Bowden and Ridler (2), using two metals which rubbed against one another and acted as a thermocouple, measured their surface temperature. By applying a flat disk of mild steel and a cylinder of constantan, and loading at the rate of 170 pounds per square inch at a surface speed of 36 feet per second, a temperature of 1000 degrees Cent. (1832 degrees Fahr.) was noted. In the presence of a lubricant with a surface speed of 32 feet per second and the same loading

This article is based on the author's booklet "Wear—A Discussion of the Mechanism of Wear Phenomena and Influencing Factors", recently published by The Nitralloy Corp.

*References in parentheses are listed at end of article.

of 170 pounds per square inch, a temperature of 600 degrees Cent. (1112 degrees Fahr.) was noted.

It would appear that up to about 400 degrees Fahr. the temperature does not materially affect wear of ferrous alloys, but it does affect the medium in which wear takes place, especially the oil lubricant. Increased temperature thins the lubricant, oxidizes the wearing surfaces, and may even temper hard materials. When temperatures exceed 400 degrees Fahr., coking of the oil may occur, while maintenance of a lubricant film becomes difficult.

Molecular Adhesion: It has been shown (3) that the attractive force of surface atoms is exceedingly powerful for a distance of 1 to 3 Angstrom units (1 A.U. $= 10^{-10}$ meters). When surfaces approach within the range of these powerful forces they seize or weld. As a matter of fact, if the surfaces of bearings approached as close as 2 to 3 Angstrom units, disaster would be encountered more often than now. Fortunately the presence of foreign matter, as thin films of oxide, and even humidity, prevent this close wedlock of two metallic surfaces. Although the attractive molecular forces act on the foreign substance as they do on similar matter, the adhesion between dissimilar molecules is much less, so that the bond is never really as intimate as that required to cause molecular cohesion. Yet it is believed that atomic attractive forces enter into the mechanism of wear and are the cause in part, at least, of galling. This would lead to the conclusion that if the surfaces were extremely hard the chance of seizing and galling would be reduced, which is partly borne out by experience.

Specific Pressures, Lubricant and Wear: Jominy (4) studied the effect of lubricant and wear on cast iron. Specimen *A*, *Fig.* 2, was run on a wear test machine by gradually increasing the load on the same bar of cast iron. When a specific pressure of 72 pounds per square inch was reached with no lubricant, the load was run for 10 minutes. Specimen *B* was run by gradually increasing the load to 4800 pounds per square inch, and operated for 17½ hours under this load using a light lubricant. The rate of wear of the nonlubricated specimen *A* was 260

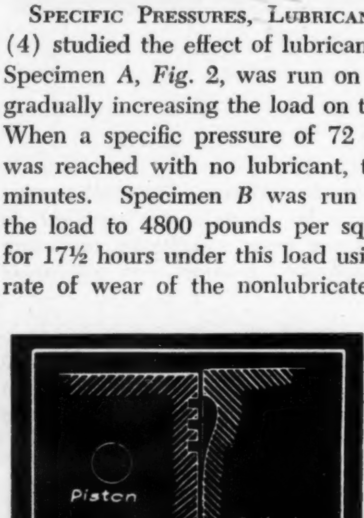


Fig. 3—Left—Greatest wear on engine cylinders occurs opposite top ring when piston is in top position

Fig. 4 — Right — Successive events during the action of the oil wedge in enlarging a fatigue crack, according to Way's hypothesis

times faster than on *B* in spite of the fact that the load on *B* was 65 times greater than on *A*. The resulting surface effect is easily seen in the two parts. When extreme pressure-lubricants are added to the oil, it is claimed that loads ten times greater, or even more, can be carried by steel surfaces without scoring. Such addition agents are employed in auto differentials because of high pressures.

Wear Phenomena: Since galling occurs due to welding and fusion, certain methods suggest themselves as a means of preventing galling. The first and the most logical would be, of course, to use mated materials which have no tendency to weld as, for example, steel and lead. The use of materials with widely separated melting points does not necessarily give assurance against seizure however, for if the lower melting material "wets" the other, welding may still occur.

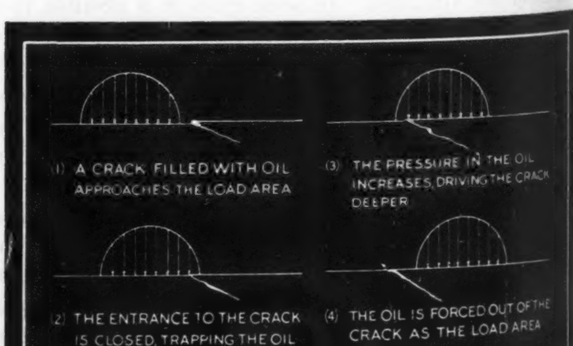
Another method of preventing galling consists of introducing a third agent or substance preventing metal-to-metal contact between the rubbing surfaces. For example, we may introduce graphite or treat the surfaces chemically. To overcome galling in cast iron various chemical processes have been employed, a few of which are: Ferrotox, Ferrox, Bonderizing and Granadizing. Granadizing has been extensively used for piston rings, valve tappets and cam shafts.

Ideal Combination To Resist Galling

Experiments have shown that an ideal combination to resist galling would be two high melting temperature metals having a reasonably separated melting-point difference, provided the metals are immiscible in the molten state and the one with the lower melting point has a high thermal conductivity and good chemical stability at high temperature. Such a pair of metals is steel and silver. As a matter of fact this combination has been used for bearings in aircraft engines for a long time. This combination has been superseded by plating the silver surface with indium.

Scoring, or pitting in spiral bevel gears, according to Almen and Boegehold (5), does not occur when the product of the tooth pressure in pounds per square inch times the velocity of travel of the tooth in feet per second is 1,500,000 or less; however, when the product of these two values is 1,800,000 or over, scoring is quite common when mineral oil is used. This assumes a file-hard surface for both gears.

Fretting: A form of corrosion-wear believed by some to be due to internal friction activated by vibration, and usually found under a press-fitted part where high stresses exist, has been called fretting. It may occur when the assembled parts are or are not in visible motion (6). There must, however, be a small amount of movement between the surfaces, and this small play gives rise to galling and tearing of tiny particles which oxidize, resulting in rust. The English named this type of wear "fretting corrosion".



Sachs called it "chafing" (7). Fretting occurs at surfaces which are considered practically fixed in relation to one another as press fits, such as a ball race in a housing, a keyed-on gear on a shaft, etc. Oil does not prevent fretting corrosion even if it modifies the effect.

Fretting trouble has been found on knuckle-pin bores of master connecting rods of the solid or one-piece type, the same malady extending to the knuckle pins themselves. Even force and shrink-fitted surfaces are subject to this form of wear. On dismantling a surface in which fretting has taken place, the shapes of the damaged areas of the parts are found to be almost identical, and a brown slime composed of oxide debris and lubricant often is found at the junction of the surfaces. When fretting takes place in aluminum or magnesium alloys, the oxides usually appear as a dark powder. The production of this powder is evidence that surface roughening and galling has occurred. To prevent galling and to distribute the clamping stresses a fiber, leather, or even a paper bushing frequently is used where such can be applied. Better still, a soft metal which itself is not oxidizable, as, for example, silver, may be electroplated on the surface, and this has sometimes eliminated the trouble.

Fretting Starts with Slip

The curious fact concerning fretting is that its origin seems to be due to a slight motion, since no fretting occurs when the part is absolutely at rest. The effect reduces itself finally to wear, and this particular form of wear commences its cycle by vibration. Strange to say, however, neither vibration alone nor alternating stresses will cause this sort of wear destruction. To produce fretting phenomena, it seems necessary to have a surface-sliding or slip; this slip is the essential starting point.

It is claimed that fretting corrosion has been observed in fittings subjected to pressures from as low as 6000 pounds per square inch to as much as 52,000 pounds per square inch. Fretting-wear results from exceedingly small amounts of slip of the order of 5×10^{-8} or 5×10^{-9} inches (.05 to .005 micro-inches). Such displacements are from a maximum of four to a minimum of less than one-half times the atomic dimensions.

Chemical Effects: It is known that the resistance to wear may be modified, for better or worse, by chemical reactions occurring in the medium in which the frictional part operates. These chemical reactions may be induced by the temperature rise of the part due to the friction resulting from its motion.

The atmosphere or medium in which the rubbing parts operate also will determine their rate of wear. In some tests made by Jominy (4), carbon dioxide was bubbled through hot water and released in front of a wear test specimen. He reports that wear was accelerated to eight times as much as when no carbon dioxide was used. Even when the specimen was copiously lubricated with oil, decided rusting occurred.

Schottky and Hiltenkamp (8) found that steel may be nitrided by the action of the air and produce a brittle layer which breaks off, thus increasing the wear. On the other hand, it has been found that an oxide formed on steel was a protection against wear (9). S. Masuhiro (10) showed that a thin film formed by cold working and oxidation improved the wear resistance of cast iron. The effect of high temperatures which facilitate oxidation will have a powerful influence on wear, for better or worse, depending on the properties of the oxide, etc., formed.

Cylinder and Piston Wear: This form of wear has been given more consideration perhaps than any other two things in automotive engineering. *Fig.* 3 shows the usual wear on cylinders. The greatest wear occurs at the top end of the piston travel under the topmost ring, and decreases from there down. It was shown by Roensch, in a paper before the S.A.E. in January, 1937, that this type of cylinder wear could be ascribed to three causes, namely: Abrasion due to foreign matter in the oil film; erosion due to metal-to-metal contact between the cylinder wall on the one hand, and the piston and rings on the other; and corrosion which results from chemical action on the cylinder walls by the products of combustion. The order of importance of the three causes varies with the condition of the operation.

The most extensive research on cylinder wear was carried out by Williams (11). The tests were made on a single-cylinder engine in order to isolate the conditions which produce wear. If the engine operated with cold jacket-water, wear at once increased. Operating at 50 degrees Cent. he found the wear was eight times that when the water was 100 degrees Cent. This led him to conclude that cylinder wear was largely a question of corrosion so that, when the cooling water temperature was low enough, the moisture formed by the combustion of the fuel condensed on the cylinder walls, and as this moisture contained carbon dioxide, and probably some sulphur dioxide, the cylinder wall was corroded. The products of corrosion thus produced were scraped off by the piston ring, so that they were not found on the wall when the engine was torn down.

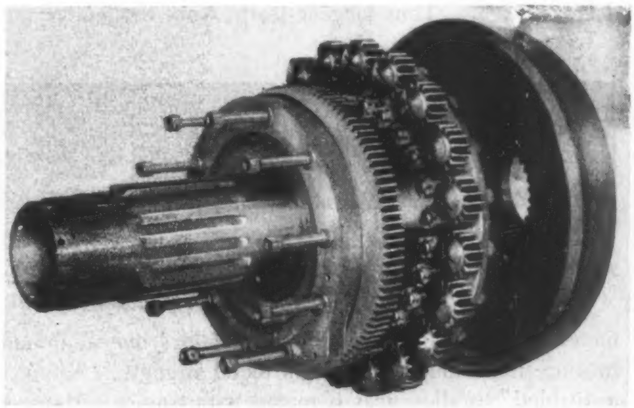


Fig. 5—Radial aircraft engine reduction gear employs nitrided ring gear and carburized pinions and sun gear. Each pinion transmits 80 horsepower at take-off, with tooth contact pressure 120,000 pounds per square inch

Williams proved his explanation of the cause of wear by driving an engine without firing, and running through it a mixture of air, steam and carbon dioxide. Under these conditions he found wear similar to that obtained by firing the engine. A rather peculiar result is this: An attempt to make cylinders and piston rings from stainless materials has not decreased the wear appreciably.

While stainless materials have not greatly changed the

wear of cylinder liners, experience has shown that increasing the hardness of cylinder walls increases the wear resistance. The General Motors corporation has found that a cylinder wall hardness of 500 brinell or more has given in all cases increased wear resistance as compared to that obtained with softer cylinder irons (4).

The Forging and Casting corporation reports that Nitricastiron cylinder liners with brinell hardness of 1000 have shown wear of only .0018-inch after 110,000 miles of service. It also is claimed that such liners will have an average life of from 300,000 to 400,000 miles.

Nitrided Nitralloy cylinders, delivering over 130 brake horsepower per cylinder, are in use on many of the biggest radial air-cooled engines. Their remarkable resistance to corrosion from the acid-bearing moisture accounts, among other characteristics, for their extreme longevity. These cylinders, shown in *Fig.* 1, are nitrided and have a surface hardness of 900-1000 brinell, and a core strength of probably 125,000 pounds per square inch.

Work Hardening: It is a curious anomaly that work hardening encountered in the phenomena of wear helps to decrease wear. Thus in gear teeth, work hardening may

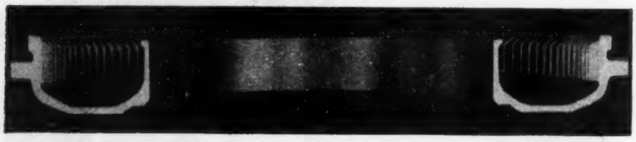


Fig. 6—Thin sections make heat treatment difficult for plate type cam and driving gear on radial aircraft engine

increase the tensile strength of soft alloys like aluminum to more than double the virgin metal strength. Again, if a nitrided Nitralloy gear is mated with, say, a carburized gear, it has been found that this increases the life of the carburized gear. In fact, in many instances, the stresses on such cold-worked carburized gears may be increased to higher values than the ultimate strength of the material in its original condition.

Gear Pitting: Gear pitting has been studied by many engineers, research institutions, and colleges. The work which is under investigation by Dr. Stewart Way of the Westinghouse Research Laboratories (12, 13), promises much in explaining the cause of gear-pitting phenomena. Briefly this report seems to show that pitting results from the formation and growth of cracks. These cracks begin as surface layer cracks, the thickness of the layer being less than .001-inch.

Pitting, it is shown, cannot be produced without the presence of oil, and if oil is added to a pair of rollers that have been run dry at a load about the pitting point for several million cycles, pitting will occur in a few hundred thousand additional revolutions. Here is the explanation given by Dr. Way as to how pitting starts and continues:

"Stresses near the surface irregularities may give rise to fatigue cracks in an extremely thin surface layer. Presumably these cracks would start in rollers run without oil as well as those run in oil. We have to explain why they suddenly begin to grow in an oblique direction when oil is added, and why this direction is always the same with respect to the direction of rolling.

"These events can be explained if we suppose that the primary failure consists of very small cracks, either horizontal beneath the surface, or inclined to the surface, and that the entrance of the oil into the cracks is responsible for their growth.

"Consider what happens when the oil enters an oblique crack. The series of events is pictured in *Fig.* 4. The extremely high pressure developed in the oil in the crack probably would result in the crack being driven deeper into the metal.

"The condition for the growth of a crack on this hypothesis is that it slopes in the same direction as the direction of rolling, which is precisely the direction taken by the pitting cracks. A crack sloping in the opposite direction would simply have the oil forced out of it as it passed under the contact area without developing any high tensile stress at the end of the crack."

Gears for Aircraft Engines: In an article on the "Manufacture of Aircraft Gears" (14), some interesting data are furnished on Nitralloy gears used on aircraft engines. One of the points brought to light is that the surface hardness of Nitralloy gears does not drop after a brief service as frequently happens with a carburized case. This observation may explain why carburized gears working against nitrided gears show increased endurance. This increase is believed to be due to the cold working of the carburized gear against the harder nitrided gears.

Nitralloy steel gears have many great advantages for production of aircraft engine gears aside from questions of high pitting resistance. One of these advantages is that its use makes possible surface hardening of the teeth of large gears and of light section which would be utterly impossible to carburize and quench, *Fig.* 6.

According to Rasmussen (15) the ability of the surfaces of gear teeth to resist surface wear is dependent upon the surface fatigue limit of the material, the respective curvatures of the mating teeth, and the relative hardness of the mating surfaces. Hertzian stresses of 350,000 pounds per square inch on nitrided gears have been reported, but the author does not believe these values are representative of the true stresses.

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Fig. 9—Gearmotor with slip clutch drives motion picture camera. Slip clutch allows threading of film without damaging worm gear

"Package" Motor-Control Units Facilitate Design

By William H. Fromm
The Dumore Co.

Part II—Motion Modification, Protection and Indication

ACCELERATED by aircraft requirements, recent developments in built-in motor control units have made possible precise control of fractional horsepower motor operation with a package unit completely tailored to the application. These assemblies often are no larger than the motor itself had been previously. In Part I, motion control was discussed. This included built-in governors, variable-speed transmission clutches, brakes and limit switches. In the present article will be covered motion modification, protection and indication.

Gear Units: In many motor applications the speed of the basic motor does not conform to every drive requirement and must be modified, usually reduced, to meet specifications. To this end a large variety of integral speed reducers has been developed. Perhaps the simplest and commonest type of gearing used is the straight spur, which can be employed in single or multiple stages to provide ratios of from slightly more than unity to several hundred to one. These gears are extremely efficient but, when made of metal for heavy torque transmission, are rather noisy. Gears of laminated synthetics are frequently used to reduce noise. Helical gears, also, are inherently more quiet than the spur type.

For equipment where higher ratios are required such as in *Fig. 9*, worm gearing is employed. With two stages it is possible to obtain ratios up to

1000 to 1, or even higher. Worm gears are quiet but relatively inefficient and usually are used only when heavy torques are not involved. The axial thrust produced by the interaction of the worm and gear must be absorbed by end thrust bearings, *Fig.* 10. Output shaft of a worm gear unit may be either parallel to or at right angles with the input shaft.

Planetary gearing is the third of the trio of most familiar types. Extremely high ratios can be achieved within a relatively small space and efficiencies are high. This

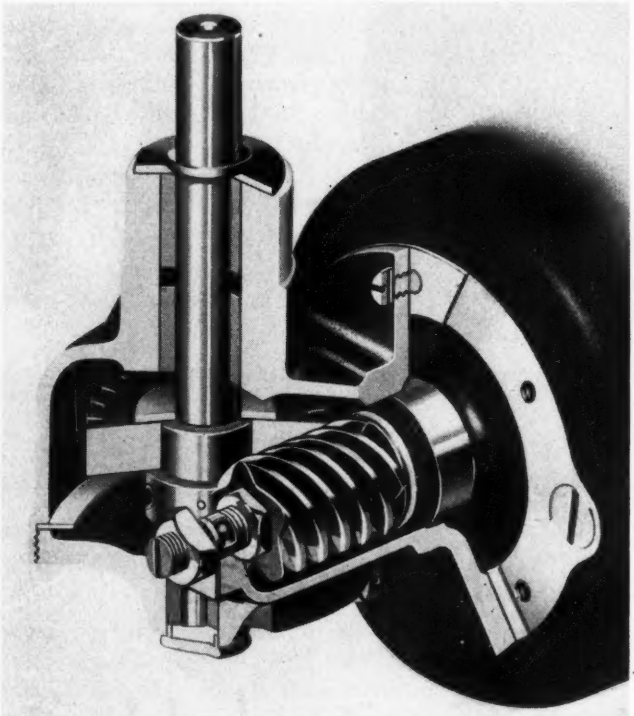


Fig. 10—Cutaway of single-reduction worm-gear unit with thrust bearing and phenolic gear for quiet operation

type of gearing, however, is fairly expensive. The output shaft is usually parallel to and concentric with the input shaft.

There are many other gear types such as bevel gears, heliocentric gears and herringbone gears, each with its own particular function or advantage. Their common purpose is to modify the motion of the motor so it can be employed to best advantage. Use on motors of all sizes has been growing and will doubtless spread. Gearing has been particularly useful in connection with small aircraft motors where the armature speed is raised to high levels to afford weight reduction. This experience has established certain optimum speeds, approximately in the range of 7500 to 10,000 revolutions per minute, beyond which the weight of the gearing exceeds any saving which may be created by increasing the speed of the armature.

Actuators: Where machines require linear motion a means must be found to translate the rotary motion of the motor. The simplest means, widely used on aircraft equipment, is the screwjack, *Figs.* 11 and 12. It consists essentially of a long screw, keyed to prevent it from turning, which is driven by a rotating nut. A converse arrangement is a rotating screw driving a travelling nut. Tremendous forces can be obtained through screwjacks from relatively small motors. The speed of movement is determined by the lead of the threads.

The crank is another kind of motion modifier. Linear motion can be produced readily by mounting a bell crank on the output shaft of a high-ratio motor-reduction unit rotating through an arc of something less than 360 degrees. The length of travel is necessarily more limited than that of the screwjack. Furthermore the amount of gearing necessary to produce the requisite slow motion of the gear shaft usually requires more space and is heavier than an equivalent screwjack. Simpler construction and, perhaps, lower cost are the significant advantages of the bell crank.

On aircraft equipment such as cowl flaps, retractable landing gear and bomb doors where heavy pressures and short travel distances are involved the efficacy of such actuators is obvious. Machine designers will recognize corresponding requirements in such devices as door operators, valves and a host of special machines. The possibilities of linear-actuator applications will reach into almost every field of design.

The actuators mentioned in the foregoing are merely representative of the great variety of motion modifiers available and in development to satisfy the exact needs of particular machines.

Protection

Because replacement of a motor involves trouble, inconvenience and expense, there naturally has grown a strong demand for protecting motors from abuse, unex-

Photo, courtesy Lockheed Aircraft Corp.

Fig. 11—Cowl flap actuator utilizes series motor and screw jack to obtain linear motion

pected overloads, and jams due to failure of other parts. This demand is particularly strong in connection with aircraft applications where the failure of a motor endangers or prevents continued flight. Should gun fire damage a mechanism on military aircraft and cause it to jam it is important that the motor be protected against burning out when it attempts to operate and stalls. Several types of remarkably compact protective devices are incorporated in

current designs as built-in motor features, greatly increasing the scope of their application.

FUSES: The simplest type of protection is the fuse, but the usefulness of this is of an emergency nature since fuses have very slight thermal or time-delay characteristics. Frequently, however, they are used in conjunction with thermal protectors to prevent burnout of heater coils. Fuses may be installed in accessible portions of motors but are more generally connected remotely for convenience in replacement.

THERMAL PROTECTORS: One of the surest methods of preventing burnouts is to have a temperature-sensitive element, *Fig.* 13, built into the motor and to cause it to actuate a switch in the motor circuit. An application of this type is illustrated in *Fig.* 14. Such elements ordinarily are bimetallic pieces which bend or snap as the temperature changes, opening and closing the circuit according to the direction of change. In addition to sensing the temperature of the motor these protectors frequently have a heater coil carrying motor current, so that the bimetal will respond quickly to stalls or heavy overloads and more slowly to lighter overloads. Should the motor current increase with increased load, and the temperature of the motor frame and of the heater coil rise to dangerous values, the bimetal moves far enough to open the motor circuit. Sufficient time delay is inherent in the bimetallic element to prevent tripping on high inrush starting currents.

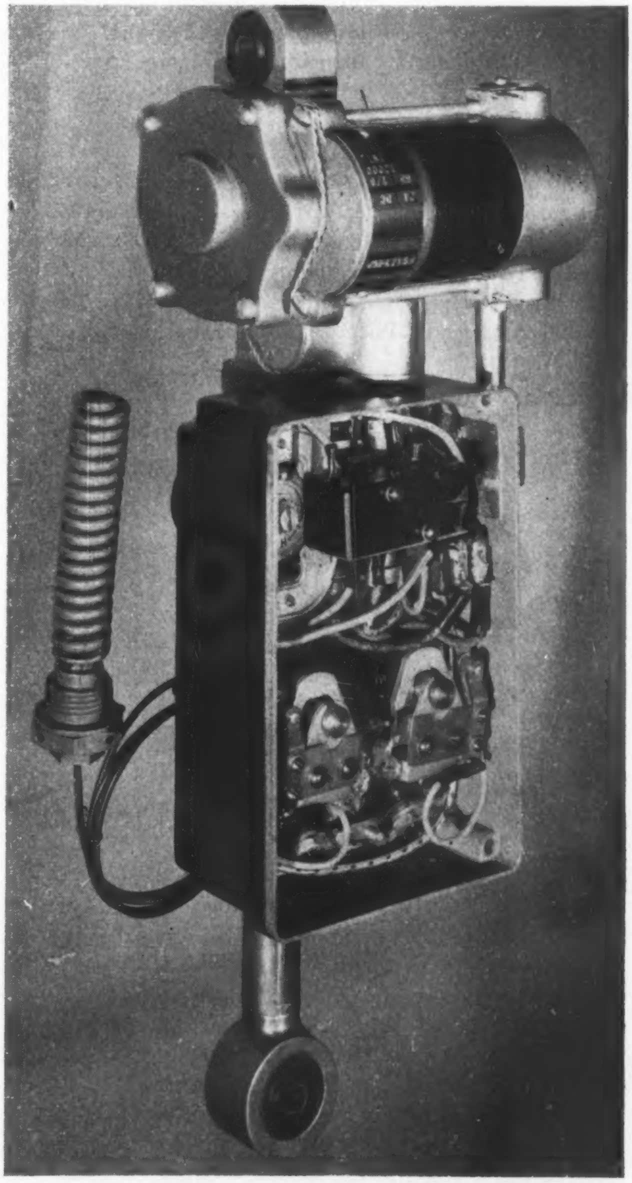
Thermal protectors are either the manual or automatic reset type, the former keeping the motor off the line until the operator presses the reset button, and the latter closing the motor circuit as soon as the temperature has dropped to safe levels. Both have their uses and selection is determined by whether or not, in the interests of safety or similar considerations, it is desired to exercise supervision over restarting. One small protector widely used in aircraft motors makes use of a bimetallic disk which is concave in one direction when cold and snaps to the opposite position when hot. The disk itself is used as one of the switch contacts.

The design of bimetallic elements and heater coils must be carefully based on the requirements of each application. Consideration must be given to such factors as frequency of operation, ambient temperatures, amount of safety factor which can be designed into the motor, and importance of avoiding unnecessary tripping. Surprisingly stringent requirements can be met. For example, protectors on aircraft motors must function properly in temperatures ranging from -65 degrees to 160 degrees and in many cases must protect a motor stalled continuously for several hours.

Thermal protectors on postwar appliance motors will greatly increase sales appeal. Upkeep costs will be substantially reduced by the prevention of burnouts which otherwise might be caused by failure of other parts of the device. The prewar use of protectors on refrigerator, washing machine and oil burner motors unquestionably will expand to a wider postwar use of protector devices on many other machines where temperature may be used to initiate a control function.

SLIP CLUTCHES: A mechanical protector, the friction clutch is designed to slip at a predetermined torque. Normally it consists of two plates held together by a spring. The amount of spring tension, which can be made adjustable, determines the maximum torque which can be transmitted before the plates slip to limit the load a motor will carry and hence the current drawn. By the same token this prevents excessive torques from being applied to the driven machine and thus protects it from damage. Slip clutch disadvantages are wear, spring fatigue (and consequent change in setting) inability to protect the motor from light overloads of long duration, and lack of thermal sensitivity.

Slip clutches have been applied effectively in military equipment and will have practical benefit to offer postwar machines. One outstanding application is its use in a



Photo, courtesy Penn Switch Co.

Fig. 12—Control for aircraft screw-jack actuator has thermal element to control position of unit

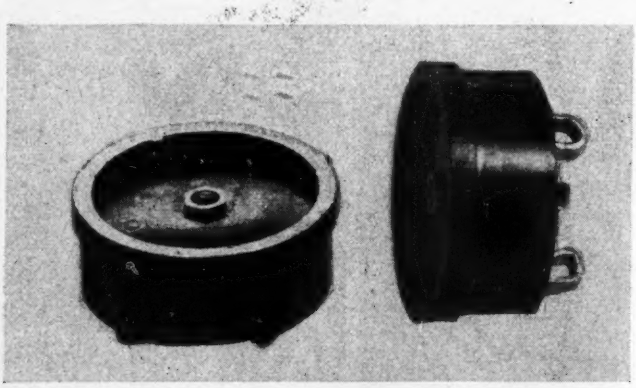
small motion picture camera motor, *Fig.* 1, primarily to protect the delicate parts of the camera mechanism. In another case slip clutches have been applied effectively to prevent shock loads on motors from driving up against

positive stops. Analysis of the potentialities of the slip clutch will reveal many other methods of utilization.

Indication

Position Indication: On certain types of applications where the driven member must be moved to various positions, it is desirable or even essential that the operator have constant knowledge of the exact position. If the driven member is out of sight some sort of indicator is needed; prime examples at present are the cowl flaps and oil-cooler exit flaps of the modern military airplane. The position of these flaps determines the amount of cooling air passing over the engine or through the oil cooler. Knowing that a certain setting of the flaps is correct for a given set of flying conditions the pilot can establish that setting by pressing the pushbutton which controls the flap motor and by keeping the motors in operation until his instrument panel indication shows that proper flap position has been reached.

Motors are now being produced with built-in means of indication, and position may be mechanically or electrically indicated. Perhaps the simplest mechanical means is a take-off shaft geared to the output shaft and connected to a dial indicator by a flexible shaft. Obviously there are practical limitations on the distance between motor and indicator, and a mechanical system places additional frictional load on the motor as well, particularly at low temperatures. The more flexible and lighter of the two systems is the electrical. One version has a small potentiometer or a rheostat built into the motor with the moving contact geared to the output shaft. A remotely located instrument of the ammeter type is connected in series with the rheostat, or one of the voltmeter type may be connected across the potentiometer. As the output shaft turns and the driven member moves, the current or the voltage in the indicator circuit is varied proportionally. This variation is translated into movement of the indicator needle to register mechanism position on a dial.



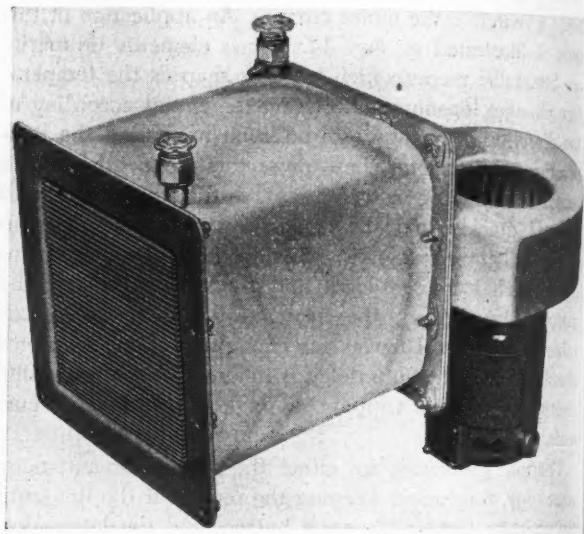
Photo, courtesy Spencer Thermostat Co.

Fig. 13—Automatic reset thermal overload unit to protect motor windings against overheating

Where motion or position indication can be used in peacetime is a question machine designers and motor manufacturers can answer cooperatively—perhaps on control equipment for ships, on valves, on furnace damper or ventilation louver actuators.

Speed Indication: Many machines, to be used to best advantage, must be operated at their optimum speeds for given conditions. The operator therefore must know to within close limits the number of revolutions per minute or feet per second or pieces per hour at which the device is performing. The widely used tachometer is now being built into the driving motor to provide continuous intelligence of operation. Professional motion picture cameras are, for obvious reasons, equipped with speed-indicating devices. Various kinds of machine tools employ this type of indication as well.

Mechanical operation of rate indicators is most practical. With the instrument mounted on the motor, direct shaft connection is possible; for remote indication a flexible shaft is suitable, with a take-off shaft provided on the motor. A tachometer installation in conjunction with rheostat control would make an excellent arrangement on a high-speed precision drill press where drilling speeds must be closely coordinated to the size of drill and type of material.



Photo, courtesy McQuay Inc.

Fig. 14—Aircraft blower unit utilizes built-in thermal overload protector

Counters: Finally, there is the type of indication which shows the number of operations or pieces produced. The simple type of mechanical counter is frequently installed as part of the motor drive. For remote indication a cam or trip on the motor actuates a lever or rod which advances the counter. The variety of applications for counters is virtually unlimited and many specific examples will suggest themselves upon brief consideration. The counting of operations or pieces can also be accomplished electrically, through photoelectric units for example; and this concept may be projected to include recording of permanent visual records.

The foregoing discussions have been concerned only with those types of accessory equipment now being incorporated in fractional horsepower motors. However, the field of application is so broad, wartime developments so prolific and the flexibility and versatility of small motors so great that the variety of controls, modifiers, protectors and indicators will inevitably and speedily be increased.

Centralized Lubrication Insures Bearing Life

By John W. Greve

Part I—Piston Valves

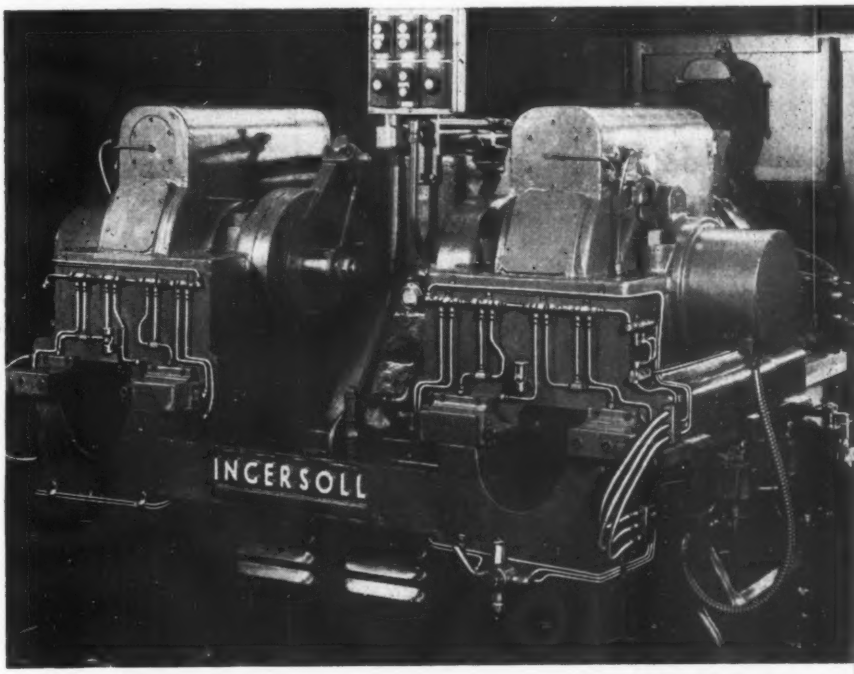
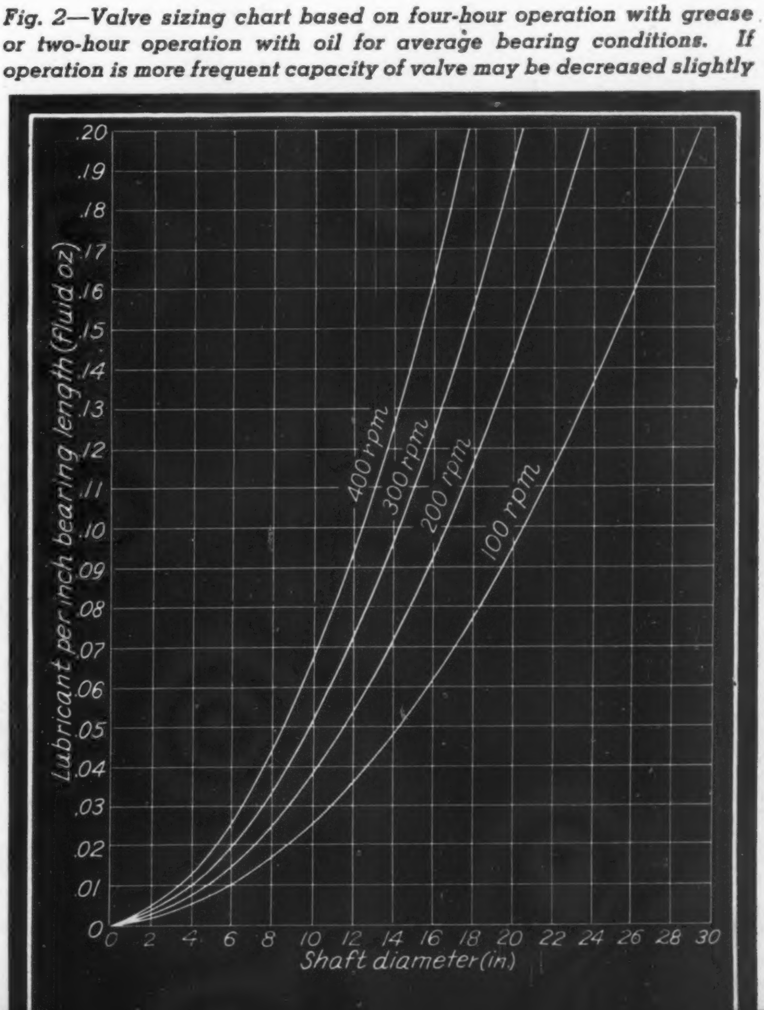


Fig. 1—Centralized system on a milling machine supplies all bearings with a measured amount of lubricant at definite intervals assuring maximum life and performance of machine

Fig. 2—Valve sizing chart based on four-hour operation with grease or two-hour operation with oil for average bearing conditions. If operation is more frequent capacity of valve may be decreased slightly



CENTRALIZED lubrication systems have become a necessary refinement for many types of machines, especially in the medium and heavy fields. Emphasis on the necessity for fast and continuous production, as well as on the high costs of shutdowns for maintenance, has added significance to the possibilities of lubrication systems for giving longer hours of uninterrupted operation. Also, today's manpower situation and overburdened maintenance crews have attracted increased attention to the advantages of automatic lubrication.

Few machines that are designed for long life would not be benefited by such a system. Three bearings would justify centralized lubrication. Where there are more bearings, where some are inaccessible, where direct manual lubrication would require a shutdown or where sufficient lubrication is absolutely essential at all times, the need for a centralized system becomes even more apparent.

Costs for centralized lubrication, including equipment and installation, usually are one or two per cent of the value of the machine. Installation cost about equals that of the equipment. In smaller machines, total costs may run as high as five per cent. Industrial equipment, bottling machines, food processing machines, cranes, textile machinery, machine tools and presses are typical of the machines that utilize centralized systems such as shown

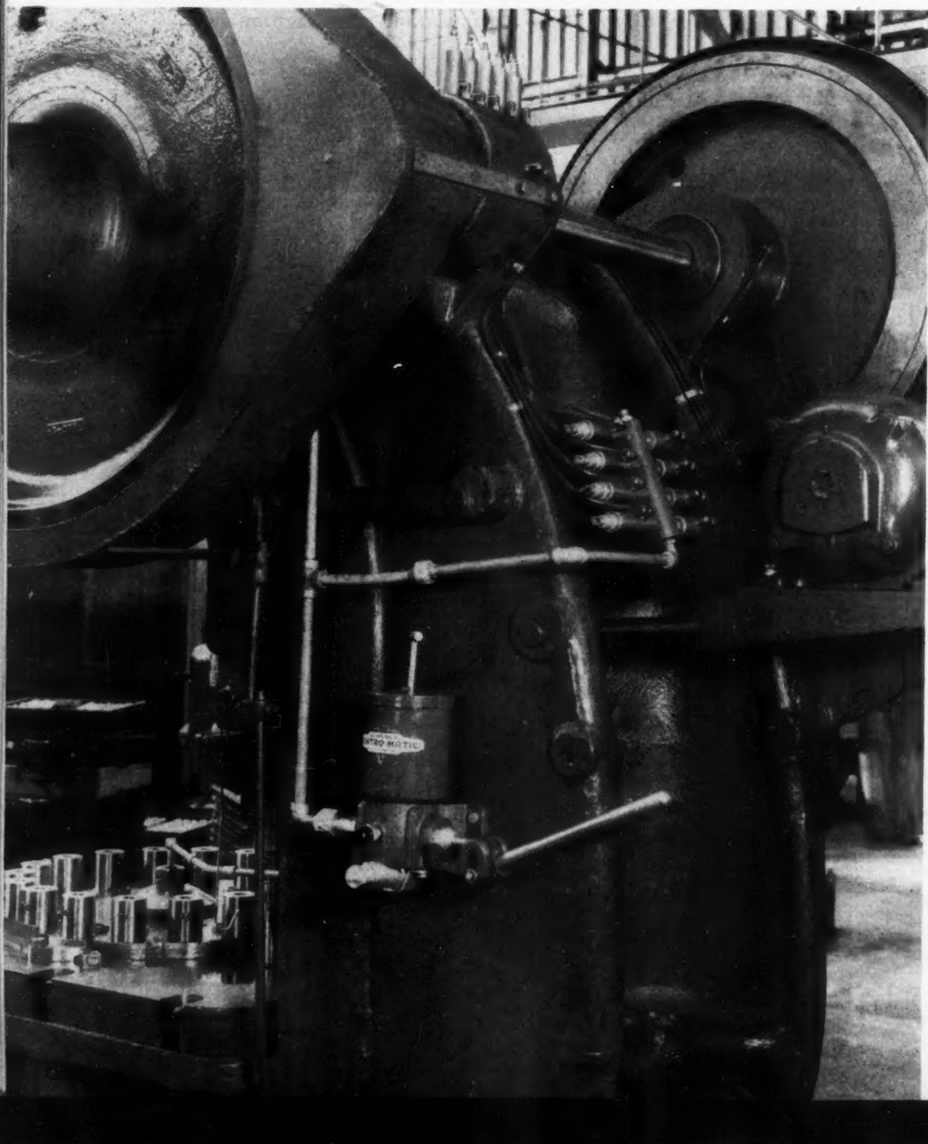
in *Fig.* 1. Consumer goods and other types of machines that are engineered to perform for a limited number of hours at an established cost in a competitive market or have limitations as to weight or size requirements, do not benefit from centralized systems. Instead it has been preferable to take precautions in the design of bearings to provide for reasonable life. Sealed-lubrication bearings and so-called oilless bearings are particularly suited to such applications. Aircraft installations also are limited chiefly because auxiliary equipment carries a high premium and because it is necessary in any case for a crew to service a plane carefully after each flight. At that time lubrication is more satisfactorily performed from ground equipment. This is also generally true of the automobile.

Delivers Definite Quantity to Each Bearing

Centralized lubrication systems as covered in this discussion include those systems for metering a definite quantity of oil or grease from a central reservoir to remote bearings at intervals, operated either manually or automatically. Circulating splash or bath systems will not be discussed although they have advantages where applicable and effect the optimum in lubrication. They are particularly useful where circulating oil may be employed to dissipate bearing heat. Such systems, however, are expensive and become prohibitive for remote bearings as far as cost is concerned. For these remote bearings, centralized lubrication systems of the types discussed in this article were developed.

Oil is the best lubricant known and may be used in all of the systems discussed. For many, oil or grease is optional. Oil is necessary for high-speed operation but is more expensive because feeding is more

Fig. 3—Manual operation for centralized system on a punch press



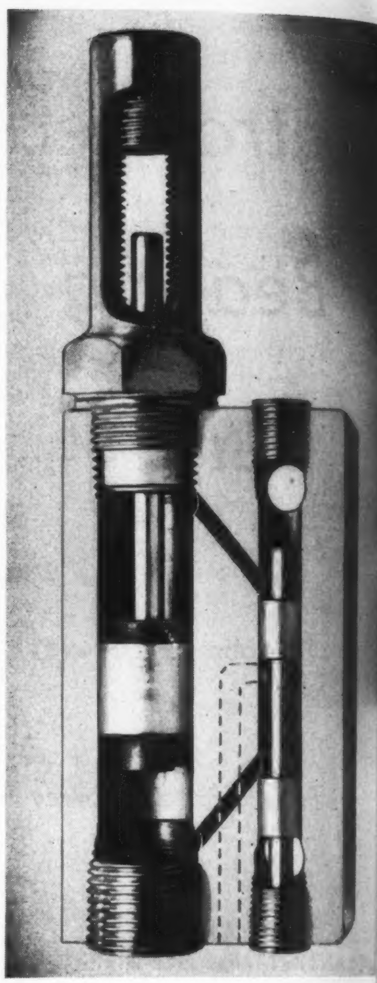


Fig. 4—Two-line type of displacement valve in which sliding spool reverses direction of valve operation

frequent. Grease, having a filler as a body for the oil, is used for many applications, however, because it is easier to control, is easier to handle, is cleaner, and excludes foreign particles from bearings by virtue of its solid nature. Oil on the other hand has an affinity for abrasive and dirt particles and might possibly carry them into a bearing after leaving the lubricator valve.

Grease remains in a bearing for long periods and, for that reason, is particularly adapted to hand systems. Normally, antifriction bearings are lubricated with grease. Protection against overloading is necessary, however, requiring a safety device to prevent overheating or blowing out seals. Such bearings should not be over two-thirds full.

Grease is necessary for elevated temperatures, but extreme temperatures necessitate the use of oil because grease would cake. For such applications, high filter oil is used to prevent carbonization. For heavy loads and slow speeds grease is preferred. Sometimes, however, it is necessary to employ graphite. The com

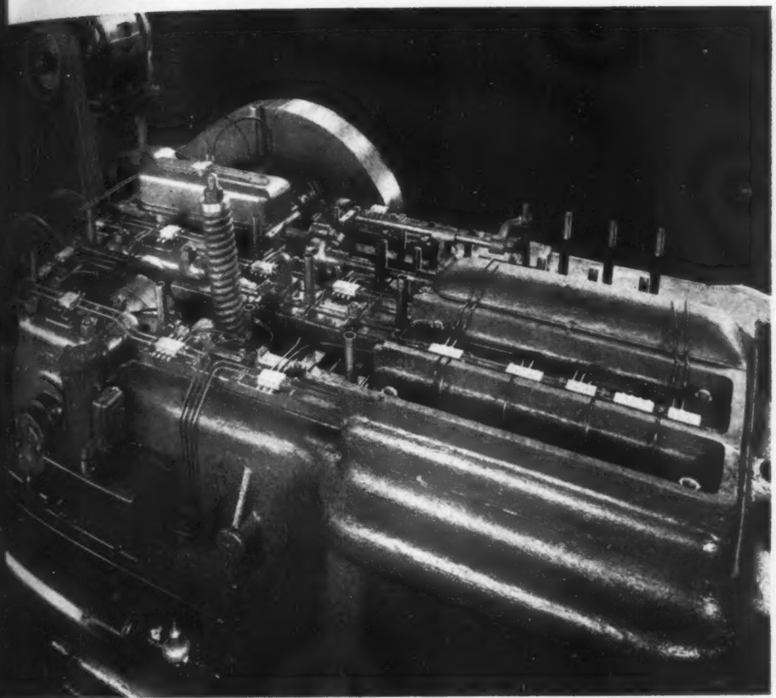


Fig. 5—Full automatic centralized lubrication applied to a forging machine. Motor-operated, time-clock controlled system lubricates fifty-five points

mercial grade of graphite flake has abrasive action harmful to lubricator parts, and premature wear of pumps would result from use of graphite unless the electrolytic grade is employed.

Quantity of lubrication per bearing depends upon the lubricant, type of bearing and kind of service as well as the frequency of application, bearing size and journal speed. Each manufacturer's valves are available in sizes to deliver the quantity of lubricant desired. After the requirements for each bearing are determined, valves and pumping capacity are selected to suit the needs. In Fig. 2 are shown average requirements for bearings under normal conditions. This chart is based on manual systems for grease lubrication every four hours of operation or for oil lubrication every two hours. If more frequent operation is desired or automatic operation is employed, one size smaller valve or possibly swo sizes smaller may be used. Quantity of lubricant shown in the chart is for each inch of bearing length.

For slides or other flat surfaces requiring lubrication, the surface area in square inches to be lubricated by each valve may be multiplied by .001 to obtain valve capacity required in fluid ounces.

In the selection of a lubrication system it is essential to apply one in keeping with the requirements of the machine. As early as possible in the design stages the system should be selected and provisions made for its incorporation. As with other commercial parts it is better to design for utilization of equipment that is standard, resulting in improved appearance and considerable reduction in cost. Fundamental requirements of any lubrication system are:

1. Dependable operation
2. Long life consistent with design of machine
3. Proper measurement of lubricant to each individual bearing
4. Capability of handling lubricant best suited to bearing requirements
5. Clean lubricant supply to bearing, free of contamination
6. Simple design consistent with application
7. Reasonable equipment and installation costs.

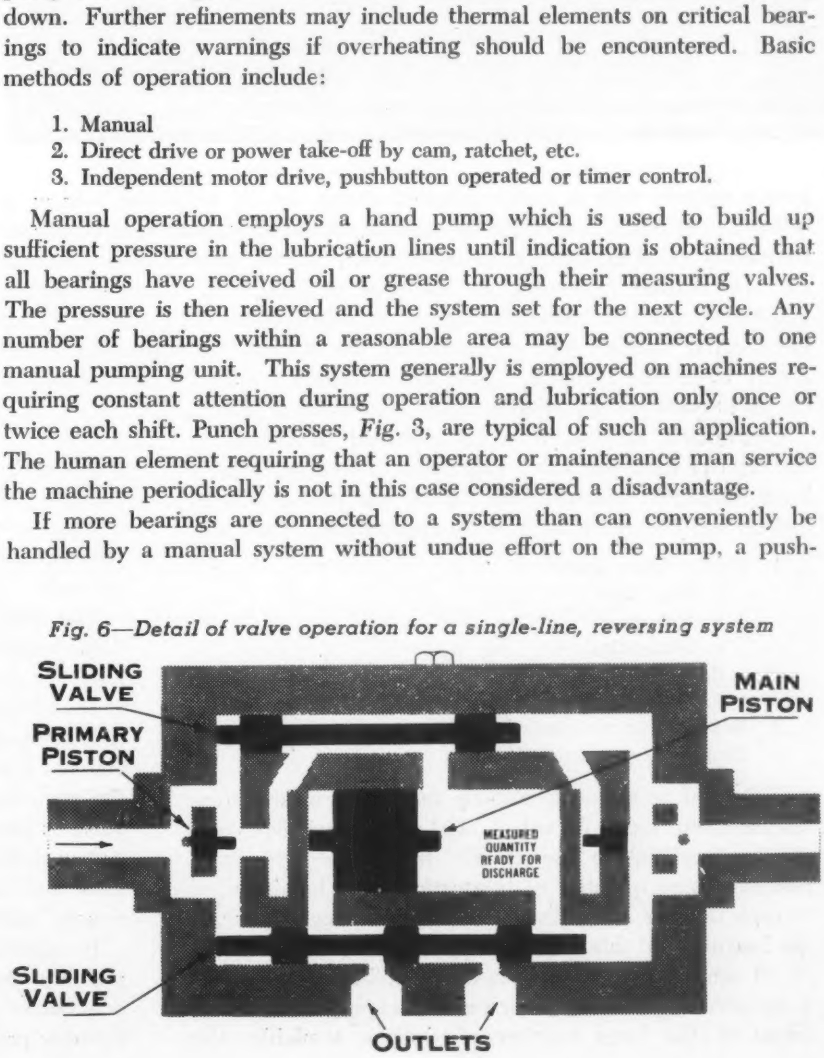
Generally the simplest design which effectively and dependably supplies proper lubrication to all bearings is the best to use. Methods of operation vary from manual pump to automatic pumps with interval timers. The latter systems may be interlocked with the operation of the machine through its electrical controls in such a manner that, if the pump does not operate and causes lubrication failure, the machine shuts down. Further refinements may include thermal elements on critical bearings to indicate warnings if overheating should be encountered. Basic methods of operation include:

1. Manual
2. Direct drive or power take-off by cam, ratchet, etc.
3. Independent motor drive, pushbutton operated or timer control.

Manual operation employs a hand pump which is used to build up sufficient pressure in the lubrication lines until indication is obtained that all bearings have received oil or grease through their measuring valves. The pressure is then relieved and the system set for the next cycle. Any number of bearings within a reasonable area may be connected to one manual pumping unit. This system generally is employed on machines requiring constant attention during operation and lubrication only once or twice each shift. Punch presses, *Fig.* 3, are typical of such an application. The human element requiring that an operator or maintenance man service the machine periodically is not in this case considered a disadvantage.

If more bearings are connected to a system than can conveniently be handled by a manual system without undue effort on the pump, a push-

Fig. 6—Detail of valve operation for a single-line, reversing system



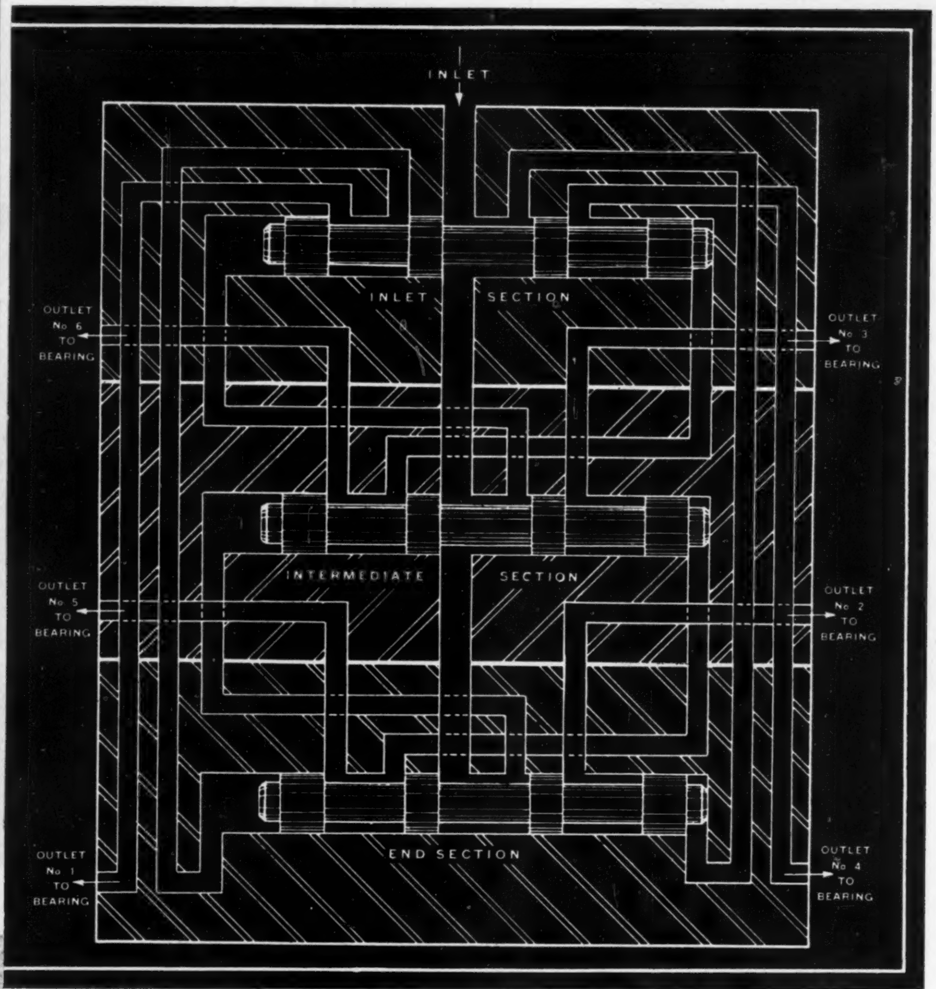
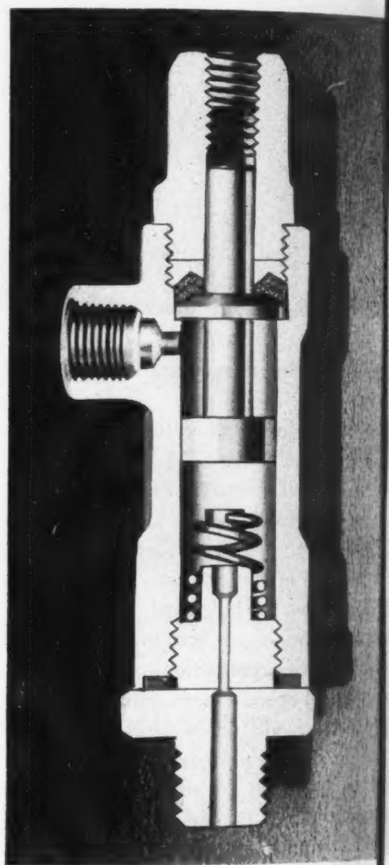


Fig. 7—Left—Proportioning type of valve serving six bearings from single line

Fig. 8—Below—Single-line type of valve which does not require reversing the pressure in the line. Operation depends on leakage past the piston and on differential areas



button system with a motor-operated pump, or an automatic system is utilized. Direct drive or power take-off for pump operation is employed frequently as an economical method. Cam, ratchet, gear or other methods may be utilized depending on the type of lubricator. Another method employing independent-motor drive in conjunction with timers, either electrical or contact type controlled by the machine, is applicable to the highly automatic systems. These are interlocked with the main machine circuit to preclude operation in event of lubrication failure.

Centralized systems may be classified according to the type of metering valve employed. Three types have been developed commercially. Each has its advantages and certain limitations for specific applications. They are:

1. Piston displacement
   a. Dual flow
   b. Reversing
   c. Mechanical
   d. Leakage path
   e. Spring return
2. Box oiler or mechanical lubricator
3. Resistance or restricted orifice.

Either oil or grease is suitable for use in most piston-displacement types of valve. The other two types are chiefly restricted to use of oil. In all the systems, a predetermined quantity of lubrication must be delivered to each bearing, regardless of the condition or location of the bearing, and this delivery must be made independently of all other bearings. Any system's ability to do this is a measure of its effectiveness for a given application. Because of the large number of designs available, this article will discuss briefly only a few types of valves to indicate the general principles involved. The discussion will not be concerned with pumps and their controls, the problems being common regardless of the type of valve utilized.

The dual-flow system, developed by Farval, is a piston-displacement type having the advantages of parallel operation for each valve with positive indication of operation at each bearing. Each valve serving one bearing has only two moving parts, a piston and spool. Quantity of oil metered is adjustable to meet varying conditions by setting the stem height on each valve. A cross section of this valve is shown in *Fig.* 4. Possible disadvantages of the system include increased pipe-line length for some applications and involved procedure in checking operation of remote bearings or a large number of bearings.

In operation the valve in *Fig.* 4 is connected to both pressure lines, one to the port above the spool and one below the spool. In the position shown, lubricant is under pressure on the top line forcing the piston down

ward by admitting pressure through the upper transverse passage. During this downward action of the piston the measured lubricant in the cylinder below the piston is passing through the lower transverse passage and through the dotted vertical passage into the bearing opening at the bottom of the valve.

The next cycle applies pressure on the lower line, moving both the spool and piston upward. This delivers the measured quantity of lubricant above the piston through the upper transverse passage and, the spool having opened the ports between this passage and the vertical passage, lubricant again is fed to the bearing. Indication of each operation is visible by movement of the stem to the extreme of its travel. Change in pressure in the lines is accomplished at the pump with a 4-way valve, either manually or automatically. Adjustment of quantity of delivery is made by positioning the stop above the stem. In locations where abrasives might be detrimental to operation of the stem and its seal, plastic shields cover the openings on the indicator.

Lubricates Every Bearing on Forger

An application of this type of valve to an Acme forging machine is shown in *Fig*. 5. This is a full-automatic system which is motor-operated and timeclock controlled. Fifty-five points are lubricated every 15 minutes with a heavy-bodied oil. The system serves the main camshaft bearings, the header slides, toggle ram slides, and die slides.

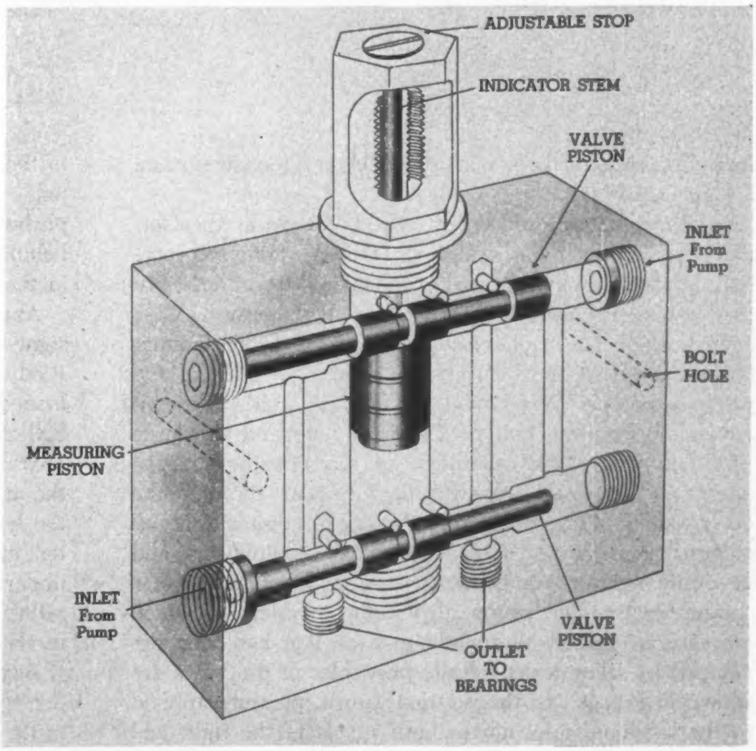
Another basic form of piston-displacement lubricator is known as the reversing, single-line system. Details of a valve for this type, designed by Trabon, are indicated in *Fig*. 6. Advantages of the system are use of a loop system in that one line is used by reversing the flow through it, indication of operation of all valves is received at the central pump, quantity of lubricant each valve delivers is preset and cannot be varied, and there are no moving external parts. Disadvantages might include increased resistance to flow of lubricant through all valves which are in series, and lack of adjustment feature to vary quantity of lubricant.

Referring to *Fig*. 6, the valve is in position for receiving pressure through the line from the left port. Left primary piston is in a floating position so that the velocity of the lubricant carries it to the right, closing its port. Lubricant advances, pushing both spool valves to the right, and admits lubricant behind the main piston. Movement of the piston delivers the measured lubricant through the right-hand outlet to the bearing. At the limit of the main piston travel a port above it is cleared, allowing the lubricant to pass through the top spool passage. This admits lubricant behind the right-hand primary piston carrying it to its extreme position and clearing the passage through the valve. This action progresses through each valve until the flow returns to the pump, indicating that each valve has operated. Flow is then reversed manually or automatically to operate each valve in reverse, completing the reverse cycle and giving indication again.

A novel valve design is shown schematically in *Fig* 7. This valve manifold is in reality a proportioning valve, designed by Trabon. A manifold of three sections serves up to six bearings. It has only three spools which proportion lubricant in sequence to the bearings. Only one line to the valve is utilized, the quantity being fed at a given rate or at given intervals. This valve is particularly adapted to automatic systems. Because in most machines bearings are grouped in various locations, cost of the lubricating equipment and its assembly is a minimum. The valve manifolds serving the bearings are grouped and are fed from larger or master valves of similar construction, completing the circuit. Conveyors or machines in which the bearings are in a long single line do not benefit by this system.

Visualization of operation of this valve without a working model is difficult, although the basic principle is illustrated in *Fig*. 7. Considering the entire system full at all times and starting the cycle in the position shown, the only passage not blocked or not balanced is from the inlet section around the right to the end-section spool. Pushing this spool to the left feeds lubricant up through the left-hand side of the inlet-section spool and to bearing No. 1. This also opens central inlet through the end-section spool to the right side of the intermediate-section spool, moving it to the right and feeding lubricant through the end-section spool to bearing No. 2. Again a port is opened at the central inlet at the intermediate-section spool, moving the inlet-section spool to the left. This feeds bearing No. 3 through the intermediate-section spool. Continuing similar spool movements the end-section, intermediate-section and inlet-section spools move back to the right, feeding

Fig. 9—Two-line, alternating-pressure system valve utilizes two spools to reverse the direction of piston movement



bearings Nos. 4, 5 and 6, respectively, completing the cycle. Continued introduction of lubricant causes the cycle to repeat.

A single-line piston displacement system which utilizes merely a piston and check valve, depending on a leakage path around the piston and differential areas for operation has been designed by Alemite recently. This valve is shown in *Fig.* 8. Piston is being forced on its way down by lubricant under pressure thus delivering a measured quantity of lubricant to a bearing through the bottom port. When piston reaches this point it closes the spring valve and continued pressure causes leakage past the piston, differential areas causing the piston to rise. As long as pressure is applied the spring valve remains shut. When piston reaches the top the pressure builds up in the line indicating that all valves in the system have operated. When pressure is relieved, spring valve opens and valve is ready for next operation.

Alemite has developed two other systems the applications of which are similar to the single-line reversing and the dual-flow systems previously discussed, although the valve designs differ in detail. An application of the single-line system, known as the dual progressive system, is shown in *Fig.* 1.

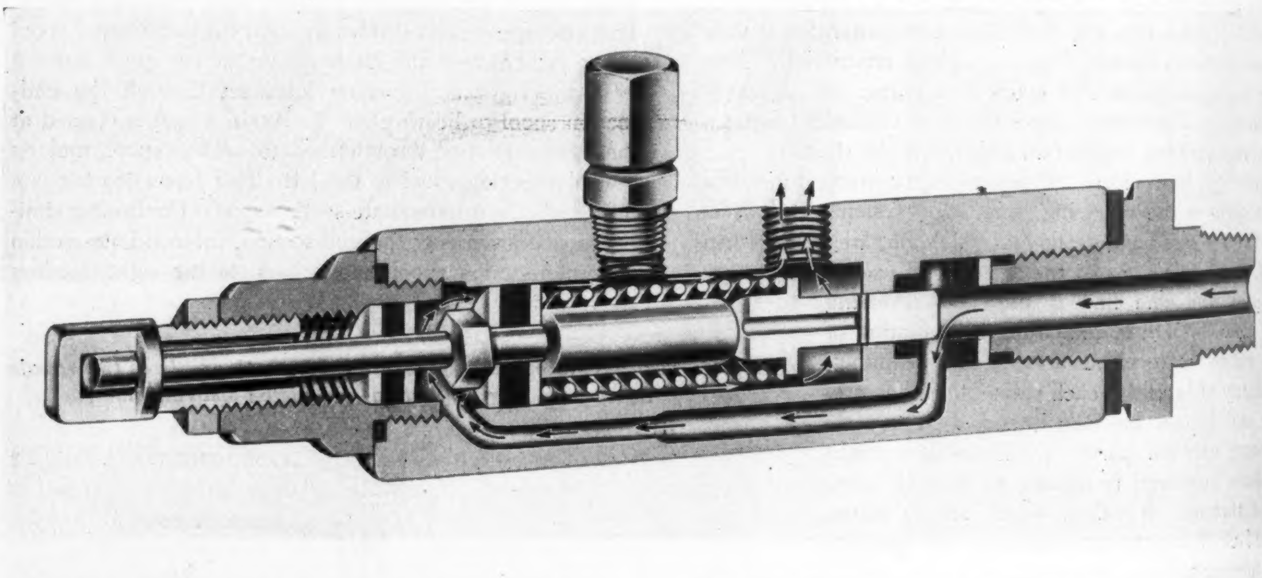
Another piston-displacement type is known as the Gordan system built by Blaw-Knox. Valve piston reciprocation is obtained by external mechanical operation. In this system the lubricant is always under pressure. With each shift of the mechanical mechanism a fixed quantity is discharged to the bearing from alternate ends of the piston chamber. When the valve is reversed, lubricant enters the opposite end of the main bore and the large piston moves to the other end of this chamber to discharge a measured quantity of lubricant through the other outlet. The mechanical linkage requires that all valves be in proper alignment which complicates and generally restricts applications to large machines.

Another two-line piston displacement system in which pressure is applied alternately to each line has been developed by Blaw-Knox. Basic principles of the valve are shown in *Fig.* 9. In the position shown, pressure applied to the left-hand inlet moves both spools to the right and the piston downward. This action forces a measured quantity of lubricant through the left-hand outlet to the bearing. Reversing the system applies pressure to the right-hand inlet to supply lubricant to the right-hand bearing outlet, completing the cycle. Delivery of lubricant is measured by an adjustable stop on the indicator stem. Position of this stem with respect to applied pressure on the line is a tell-tale of operation.

A spring-return type of piston-displacement valve, operated by single-line pressure feed, is shown in *Fig.* 10. This valve, designed by Lincoln Engineering Co. is connected in parallel to the line and depends on a spring action on the piston for resetting instead of reversal of pressure. An application of this valve is illustrated in *Fig.* 3.

Referring to *Fig.* 10, the system is full of oil. Pressure applied to the port at right has moved a slide valve to the right and uncovered a passageway. This admits lubricant to the left side of the piston, moving it to the right and delivering measured lubricant to the bearing through the port at the top. Continued pressure moves the slide valve to the right by action of a spacer between it and the piston. When pressure is relieved on the line the spring pushes the piston back to the left by moving the lubricant behind the piston through the passageway and a passage in the slide valve.

Fig. 10—Spring-return valve operates from a single line, non-reversing system. Piston is reset for next operation by spring after pressure is relieved



Another single-line, spring-return piston type valve, designed for oil delivery only, has been developed by Bowen Products Corp. The metering valve is simply a spring-loaded piston with a seat on its bottom together with a ball-check valve to prevent syphoning oil from a bearing. Pressure on the valve forces the piston down, delivering the amount of oil required. This amount is controlled by the length of the piston employed. Bottoming of the piston prevents leakage to the bearing while the system is under pressure. When pressure is relieved, the spring returns the pistons and leakage charges the valves from a reservoir for the next operation.

Box oilers or mechanical lubricators and restricted orifice types of centralized lubrication systems will be treated in the following article of this series.

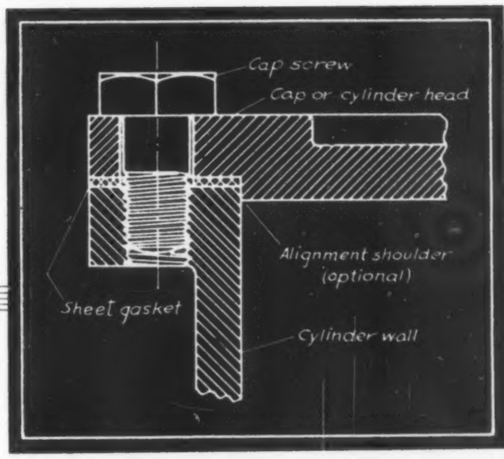


Fig. 1—Typical gasket seal for engine cylinder head. Bolts are loaded by gasket squeeze plus head pressure

Selecting Hydraulic Seals

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HOW well a given hydraulic packing installation achieves its main purpose of confining the hydraulic fluid within the components of the system is a measure of its efficiency. There are, of course, secondary considerations in packing design such as friction, life and ease of service or repair. Frequently it will be found that several different designs of hydraulic packings are equally efficient. This article will discuss briefly a number of generally accepted designs and related engineering factors.

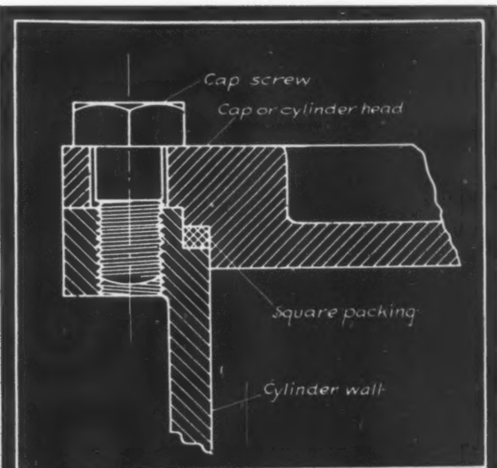
Classification of Packings

Packings fall into two major classifications: Static seals and dynamic seals. In the first group are gaskets for effecting a leakproof seal between two surfaces having no relative motion. In the second group are a seemingly endless variety of devices for preventing the flow of fluid between two surfaces having relative motion.

Dynamic seals are of two basic types. In one the motion is transverse to the seal, as in the case of a piston rod; in the other the motion is along the seal, as for a rotating shaft. Frequently the same dynamic seal design can be used for both types, and such designs may be used as a seal under a combination of transverse and rotary motion.

Static seals may also be subdivided into two major groups, one in which friction is an important factor in effecting the seal, such as in simple gaskets; the other in which the packing is confined in a chamber and increased pressure distorts the packing to effect a better seal. This distortion due to pressure jams the packing across the leakage path.

Packings have been made from many different materials such as



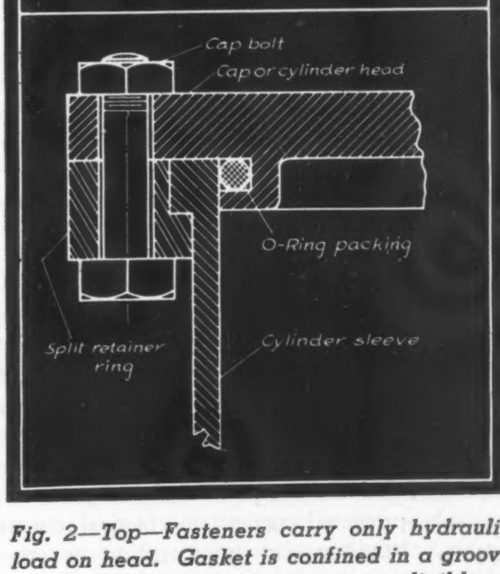


Fig. 2—Top—Fasteners carry only hydraulic load on head. Gasket is confined in a groove so that squeeze pressure is negligible

Fig. 3—Above—Packing groove in head offers an alternative to the design shown in Fig. 2

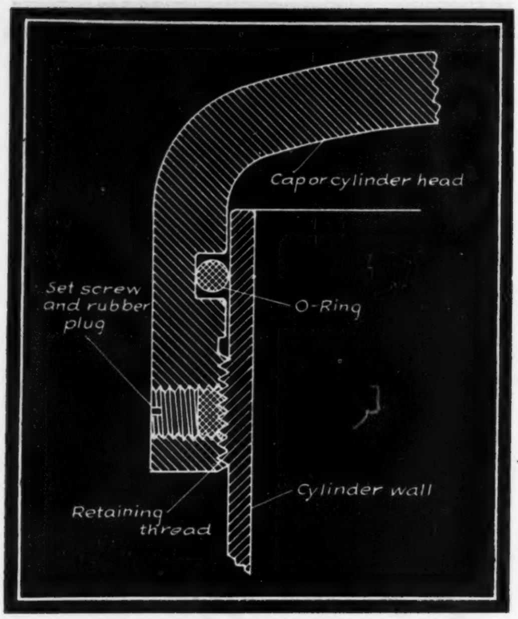
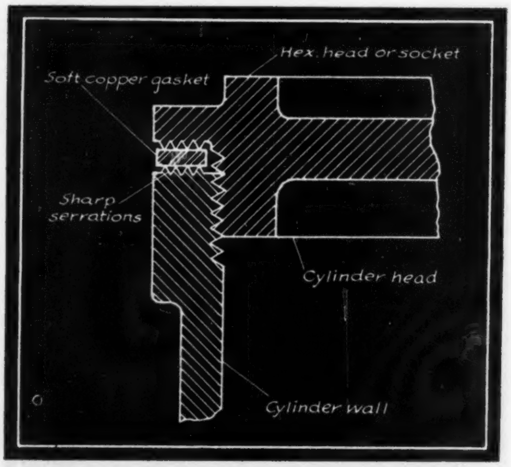


Fig. 4—Above—Threaded-sleeve construction utilizing O-ring packing

Fig. 5—Below—Copper gasket seal. This construction has concentric serrations to effect seal as head is tightened on cylinder



organic fibers, mineral fibers, natural and synthetic rubber, cork, paper, soft metals, and many other materials. Present packing trends lean toward the use of synthetic molded packings, sometimes including binders of organic or mineral fibers. It is not within the scope of this article to discuss in detail the merits of these various materials.

Importance of Surface Finish

One of the most important factors in the operation of a dynamic seal is a smooth, hard surface for the packing to slide on. It cannot be too highly stressed that mirror-like finishes increase the packing life many times. Such finishes are relatively easy to achieve on modern honing and grinding equipment. It is, of course, necessary to insure that this surface, particularly on parts in storage, be suitably protected against mechanical damage through careless handling and from rusting or other chemical changes.

The same comments apply in some degree to static seals, although the latitude is wider. Surfaces to be gasket-sealed do not need to be mirror-finished, but may have surface scratches of a minor order and depth. If surfaces to be gasket-sealed are of a high order of flatness and smoothness, only a thin paper gasket is needed to effect good sealing up to 1500 pounds per square inch.

Fastenings

In designing static seals it is important that as nearly uniform gasket pressure as possible be achieved. If the gasket is under a cover plate and this is bolted to the main structure, the spacing of the bolts or screws and the stiffness of the plate must be such that sufficient squeeze will be exerted on the gasket over its entire area. A thin cover plate will require a large number of small screws, whereas a heavy cover plate of the same area will be satisfactory with a smaller number of large screws[1].

In all machine elements using gaskets the effect of the applied hydraulic load must be considered carefully. The amount of deflection in the gasket-sealed elements must be minimized by proper distribution of metal between the fastenings. If a gasket is under considerable hydraulic load, it is better to use more care in getting a flat, rigid surface and to employ a thin gasket.

The ideal fastening for a gasket seal would be a uniformly applied squeeze over its entire area. This condition is approached, for example, in a screwed-on cylinder head with a ring-shaped gasket between the head and the end of the cylinder.

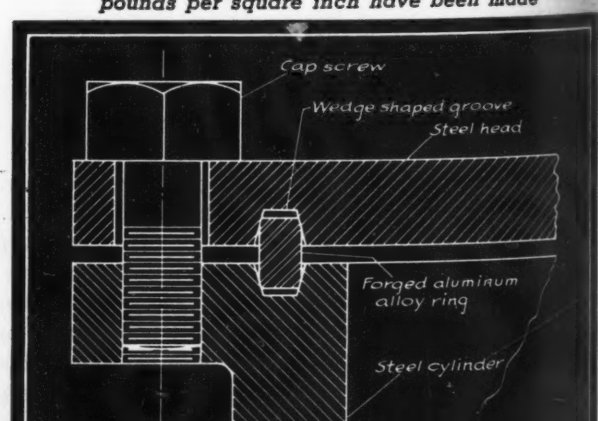
Where packing material is confined in a groove, as in *Fig.* 2, the fastenings are required to carry only the hydraulic load as applied to the cap or head—the gasket squeeze in this design being practically negligible. It is important that the fastenings be so proportioned that the cap is not allowed to separate from the cylinder wall and allow the gasket to extrude through this opening.

Effect of Fluids and Temperature

Choice of packing material and the design of the seal are influenced by the hydraulic fluid acting on the packing as well as the temperature range throughout which it must operate. Certain materials swell to a considerable extent in the presence of hydraulic fluids, and proper allowance must be made for this swelling, particularly if the packing material is closely confined.

[1] F. C. Thorn—"Gaskets", *Industrial and Engineering Chemistry*, February, [illegible], Pages 164-170.

Fig. 6—Below—High-pressure seal employs forged aluminum ring in tapered groove. Successful seals up to 20,000 pounds per square inch have been made



Packing materials should not break down under prolonged exposure to the hydraulic fluid, allowing particles of packing to drift through the hydraulic system. Wherever possible, the design of the seal should be such that a minimum of packing is exposed to the hydraulic fluid.

The choice of material is also influenced by the temperature range. Military aircraft must operate over

TABLE I

Recommended Sections for Square Packing

Head Diameter (in.)	Packing Thickness (in.)
Up to 1	3/32
1 to 2	1/8
2 to 3	5/32
3 to 5	3/16
5 and over	1/4

ranges of temperature from −65 degrees Fahr. to 165 degrees Fahr. This range is probably far greater than that found in the average industrial or commercial installation. On the other hand, many industrial applications require packings to operate at temperatures in excess of 500 degrees Fahr.

Static Seals

In *Fig.* 1 is illustrated a typical gasket seal on a conventional engine cylinder head. The head may be lined up with the cylinder by means of bolts, dowels or a shoulder. In any event, the gasket is placed between

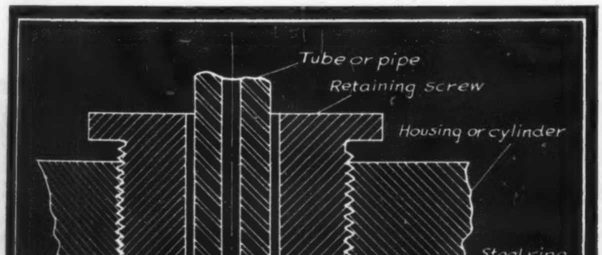
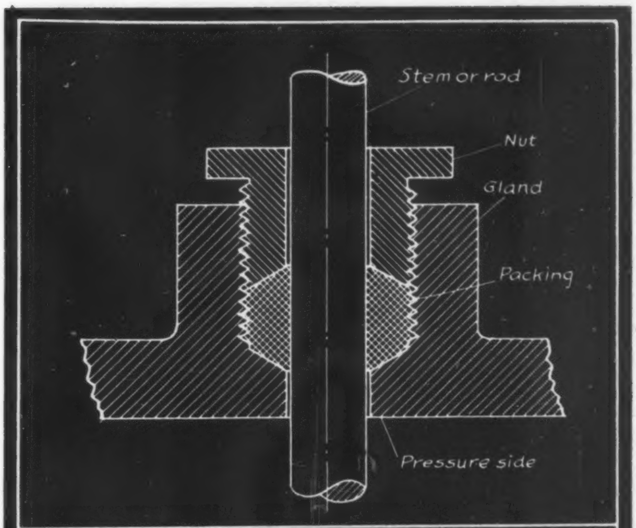


Fig. 7—Extreme-pressure packing for pipe connections, cylinder heads and pistons

the cap and the cylinder on carefully machined surfaces. Pressure from within the cylinder tends to push out the gasket against the friction holding it in place. In this design the force on the cylinder head is usually based on the area enclosed by the bolt circle. If the gasket is not pierced by the cap screws, but is a flat ring wholly inside the circle of cap screws, then the hydraulic force on the head is based on the outside diameter of the gasket. These methods of calculating cylinder-head loads are, of course,



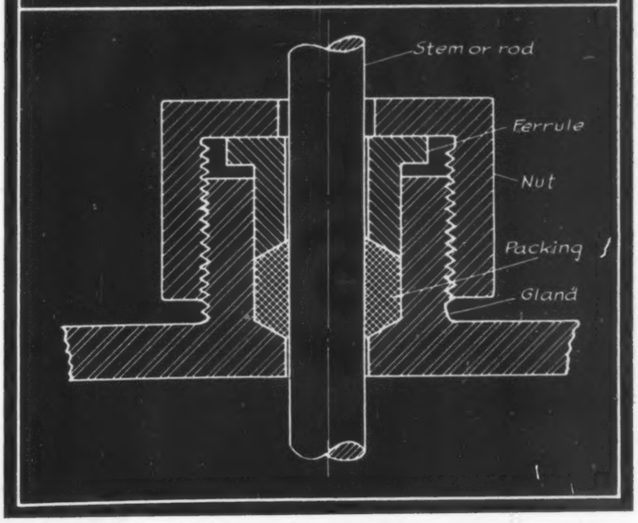


Fig. 8—Top—Common stuffing box with wick packing

Fig. 9—Above—Stuffing box with smooth gland. Nut fits on external threads

empirical but will give safe values.

The chief advantages of this type of seal are its simplicity and low cost. The disadvantage lies in the fact that the cylinder head bolts must carry the increased load of gasket squeeze plus cylinder pressure.

A more up-to-date design in which a packing groove is employed is shown in *Fig.* 2. This groove is usually placed as close to the cylinder bore as possible to reduce the total pressure load on the hold-down bolts. The groove may be in either the cylinder, *Fig.* 2, or the cap, *Fig.* 3. The packing may be of many different cross sections, but square or round packings are favored.

Square packing, if made of a compressible compound such as cork with a synthetic-rubber binder, is designed to fill the groove completely. Actually the groove has a sectional area approximately 30 per cent smaller than the packing section. The design of the cylinder and the cylinder head is such that when the packing ring is assembled between them and compressed by tightening the cap screws the packing material is entirely confined in the groove. Suggested square sections for various mean diameters are listed in TABLE I.

These sizes are based on ease of handling the rings

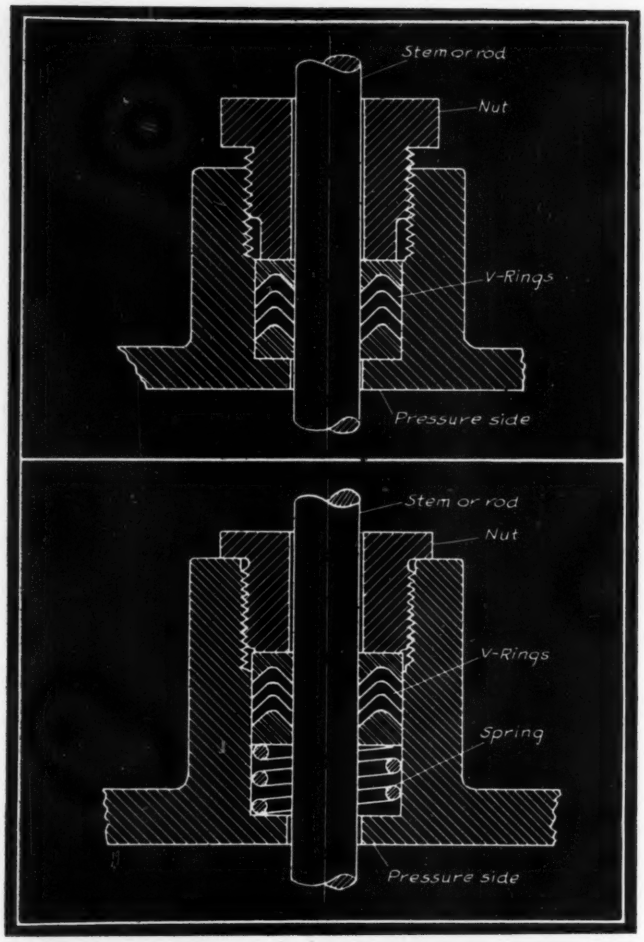


Fig. 10—Top—V-ring design reduces the friction obtained with stuffing box

Fig. 11—Above—Spring-loaded packing assures proper pressure, protecting against over-tightening

rather than on other factors. It is possible to seal a 20-inch diameter cylinder head with a ⅛-inch square section but it becomes a difficult problem to insert this section around the periphery of a 20-inch circle without stretching or twisting the packing out of shape.

To save packing material on large rings it is possible to cut long, straight pieces with the required square cross section and to wrap them around the groove, using a scarf joint without cement. The direction of the scarf does not seem to matter. This method has been tried successfully on pressures up to 300 pounds per square inch.

The round packing or O-ring, as it is commonly known, has become popular, particularly in airplane hydraulic mechanisms. When used as a static seal the section is ⅛-inch in diameter for ring sizes from ¾-inch inside diameter up to 10-inch inside diameter. This is the range in Army and Navy specifications. Larger sizes can, no doubt, be made if mold equipment is available. Engineering details for the installation of, and gland design for, O-ring packings are discussed in a later paragraph.

In *Fig.* 3 the cylinder is made with a shoulder, either integral as illustrated or brazed on. The cap is fastened to the cylinder by means of a retaining ring held or clamped by through-bolts and nuts. This retaining ring must be split in order to assemble it over the cylinder shoulder when both ends of the cylinder are identical, as is most often the case.

Cylinder-head construction utilizing O-rings and a threaded steel sleeve for the cylinder is illustrated in *Fig.* 4. This construction is typical of aircraft hydraulic design following Air Force specifications. The O-ring is held in a closely fitting groove in the cap in such a way that, when assembled with the cylinder sleeve, it is slightly compressed into the groove and against the sleeve, thereby effecting a seal. The thread serves only to hold the cap on the sleeve and plays no part in the seal. A setscrew against a rubber plug cut from an O-ring serves to lock the cap on the sleeve. Chief disadvantage of this assembly is that no appreciable torque can be transmitted from the cap to the sleeve. This is seldom necessary, so this disadvantage is of minor importance.

An all-metal gasket is illustrated in *Fig.* 5. Serrations, usually four in number, are carefully cut in the cap and cylinder so that the sharp crests of the serrations are exactly opposite each other. A carefully annealed dead-soft copper gasket is then squeezed between the serrations to effect the seal. In order to be efficient, the serra-

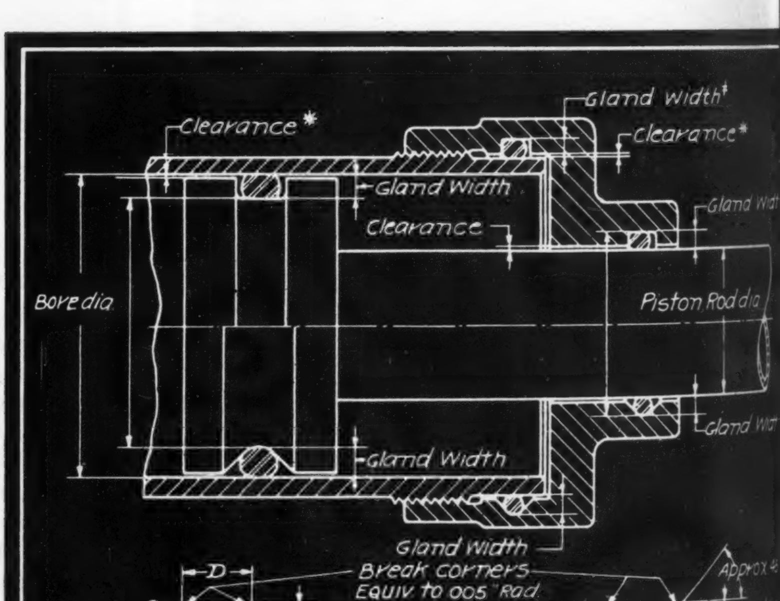


Fig. 12—Gland designs for O-ring packings for dimensions listed in TABLE *II*

tions must be smooth and hard with no "nicks" or marks, and the copper gasket must be new and of uniform hardness (or softness in this case). The copper gaskets must never be used more than once.

Allowable Pressures

The packings illustrated so far have been suitable for static seals at pressures up to 2500 to 3000 pounds per square inch. Up to 1500 pounds per square inch packings show little tendency to extrude through normal running clearances between piston and cylinder or sleeve and cap. From 1500 to 3000 more attention must be paid to keeping clearances small. At 3000 pounds per square inch extrusion difficulties are acute, unless the clearances through which the packing may extrude can be made less than .002-inch. Beyond 3000 pounds per square inch plastic or rubber packings alone have not been very successful.

Extreme-Pressure Seals

For extreme-pressure seals between a cylinder and cylinder head, a metallic seal that has been used up to 20,000 pounds per square inch with success is illustrated in *Fig.* 6. Here a forged aluminum alloy is used as the sealing means. The cylinder and head are steel. A deep groove, slightly tapered as shown and of such dimensions that the rectangular section of the aluminum packing can be wedged into the groove, is cut in both cylinder and head. On assembly the head is carefully and evenly tightened so as to bring about an even pressure on the aluminum ring, thereby insuring a sufficiently tight seal.

Illustrated in *Fig.* 7 is a type of extreme-pressure packing developed by P. W. Bridgman at Harvard University. This basic design[2] is used for pipe connections, cylinder heads, and pistons.

Dynamic Seals

It will be impossible in the space of this article to cover all the variations of dynamic seal designs, but

[2] P. W. Bridgman—*The Physics of High Pressure*, The Macmillan Co.

Table II
Gland Dimensions for O-Ring Packings of Nominal Size and AN Dash Numbers

(Dimensions Indicated on Fig. 12)

O-Ring								
	Nominal Size			Cylinder Bore ±.001 (in.)	Rod Diameter ±.001 (in.)	Gland Width Max. (in.)	Diametral Squeeze Min. (in.)	Groove Length (in.)
Dash No.*	W† (in.)	I.D. (in.)	O.D. (in.)					
1	1/16	1/8	1/4	.250	.123	.057	.010	.093
2	1/16	5/32	9/32	.281	.154	.057	.010	.093
3	1/16	3/16	5/16	.312	.185	.057	.010	.093
4	1/16	7/32	11/32	.344	.217	.057	.010	.093
5	1/16	1/4	3/8	.375	.248	.057	.010	.093
6	1/16	5/16	7/16	.4375	.310	.057	.010	.093
7	1/16	3/8	1/2	.500	.373	.057	.010	.093
8	3/32	3/8	9/16	.5625	.373	.090	.010	.140
9	3/32	7/16	5/8	.625	.435	.090	.010	.140
10	3/32	1/2	11/16	.6875	.498	.090	.010	.140
11	3/32	9/16	3/4	.750	.560	.090	.010	.140
12	3/32	5/8	13/16	.8125	.623	.090	.010	.140
13	3/32	11/16	7/8	.875	.655	.090	.010	.140
14	3/32	3/4	15/16	.9375	.745	.090	.010	.140
15	1/8	3/4	1	1.001	.747	.123	.012	.187
16	1/8	13/16	1 1/16	1.063	.809	.123	.012	.187
17	1/8	7/8	1 1/8	1.126	.872	.123	.012	.187
18	1/8	15/16	1 3/16	1.188	.934	.123	.012	.187
19	1/8	1	1 1/4	1.251	.997	.123	.012	.187
20	1/8	1 1/16	1 5/16	1.313	1.099	.123	.012	.187
21	1/8	1 1/8	1 3/8	1.376	1.122	.123	.012	.187
22	1/8	1 3/16	1 7/16	1.438	1.184	.123	.012	.187
23	1/8	1 1/4	1 1/2	1.501	1.247	.123	.012	.187
24	1/8	1 5/16	1 9/16	1.563	1.309	.123	.012	.187
25	1/8	1 3/8	1 5/8	1.626	1.372	.123	.012	.187
26	1/8	1 7/16	1 11/16	1.688	1.434	.123	.012	.187
27	1/8	1 1/2	1 3/4	1.751	1.497	.123	.012	.187
28	3/16	1 1/2	1 7/8	1.876	1.497	.1875	.017	.281
29	3/16	1 5/8	2	2.001	1.622	.1875	.017	.281
30	3/16	1 3/4	2 1/8	2.126	1.747	.1875	.017	.281
31	3/16	1 7/8	2 1/4	2.251	1.872	.1875	.017	.281
32	3/16	2	2 3/8	2.376	1.997	.1875	.017	.281
33	3/16	2 1/8	2 1/2	2.501	2.122	.1875	.017	.281
34	3/16	2 1/4	2 5/8	2.626	2.247	.1875	.017	.281
35	3/16	2 3/8	2 3/4	2.751	2.372	.1875	.017	.281
36	3/16	2 1/2	2 7/8	2.876	2.497	.1875	.017	.281
37	3/16	2 5/8	3	3.001	2.622	.1875	.017	.281
38	3/16	2 3/4	3 1/8	3.126	2.747	.1875	.017	.281
39	3/16	2 7/8	3 1/4	3.251	2.872	.1875	.017	.281
40	3/16	3	3 3/8	3.377	2.996	.1875	.017	.281
41	3/16	3 1/8	3 1/2	3.502	3.121	.1875	.017	.281
42	3/16	3 1/4	3 5/8	3.627	3.246	.1875	.017	.281
43	3/16	3 3/8	3 3/4	3.752	3.371	.1875	.017	.281
44	3/16	3 1/2	3 7/8	3.877	3.426	.1875	.017	.281
45	3/16	3 5/8	4	4.002	3.621	.1875	.017	.281
46	3/16	3 3/4	4 1/8	4.127	3.746	.1875	.017	.281
47	3/16	3 7/8	4 1/4	4.252	3.871	.1875	.017	.281
48	3/16	4	4 3/8	4.377	3.996	.1875	.017	.281
49	3/16	4 1/8	4 1/2	4.502	4.121	.1875	.017	.281
50	3/16	4 1/4	4 5/8	4.627	4.246	.1875	.017	.281
51	3/16	4 3/8	4 3/4	4.752	4.371	.1875	.017	.281
52	3/16	4 1/2	4 7/8	4.877	4.496	.1875	.017	.281
53	1/4	4 5/8	5 1/8	5.128	4.621	.240	.029	.375
54	1/4	4 3/4	5 1/4	5.253	4.746	.240	.029	.375
55	1/4	4 7/8	5 3/8	5.378	4.871	.240	.029	.375
56	1/4	5	5 1/2	5.503	4.996	.240	.029	.375
57	1/4	5 1/8	5 5/8	5.628	5.121	.240	.029	.375
58	1/4	5 1/4	5 3/4	5.753	5.246	.240	.029	.375
59	1/4	5 3/8	5 7/8	5.878	5.371	.240	.029	.375
60	1/4	5 1/2	6	6.003	5.496	.240	.029	.375
61	1/4	5 5/8	6 1/8	6.128	5.621	.240	.029	.375
62	1/4	5 3/4	6 1/4	6.253	5.746	.240	.029	.375
63	1/4	5 7/8	6 3/8	6.378	5.871	.240	.029	.375
64	1/4	6	6 1/2	6.503	5.996	.240	.029	.375
65	1/4	6 1/4	6 3/4	6.753	6.246	.240	.029	.375
66	1/4	6 1/2	7	7.003	6.496	.240	.029	.375
67	1/4	6 3/4	7 1/4	7.253	6.746	.240	.029	.375
68	1/4	7	7 1/2	7.503	6.996	.240	.029	.375
69	1/4	7 1/4	7 3/4	7.753	7.246	.240	.029	.375
70	1/4	7 1/2	8	8.003	7.496	.240	.029	.375
71	1/4	7 3/4	8 1/4	8.253	7.746	.240	.029	.375
72	1/4	8	8 1/2	8.503	7.996	.240	.029	.375
73	1/4	8 1/2	9	9.003	8.496	.240	.029	.375
74	1/4	9	9 1/2	9.503	8.996	.240	.029	.375
75	1/4	9 1/2	10	10.003	9.496	.240	.029	.375
76	1/4	10	10 1/2	10.503	9.996	.240	.029	.375
77	1/4	10 1/2	11	11.003	10.496	.240	.029	.375
78	1/4	11	11 1/2	11.503	10.996	.240	.029	.375

*Army and Navy Specification AN6227.

W† denotes width of cross-section.

the most used ones will be covered briefly. The most common is the simple stuffing box utilizing wick packing, as shown in *Fig.* 8. Here the gland has a tapered bottom and is threaded all the way to this taper. The packing consists of strands or wicks of suitable material, usually cotton or asbestos fibers impregnated with graphite, which is wrapped around the rod or shaft and "stuffed" into the gland. A suitable nut or plug serves to retain the packing and apply pressure to force it against the rod or shaft.

The stuffing gland may be a smooth hole with a smooth plug as in *Fig.* 9, pressure being applied by a nut working over an external thread on the gland and forcing the plug into the gland by means of a flange. Chief disadvantage of the stuffing-box packing lies in the fact that when sufficient pressure from tightening the nut is used to effect a seal the friction is high. Stuffing boxes can be used for both reciprocating and rotating shafts. Size depends on the type of material used for packing and, in many cases, the space available. No definite design data can be recommended.

V-Ring or Chevron Packings

High friction is overcome to a great extent by the frequently-used V-ring or chevron packing. In this packing, *Fig.* 10, a series of one or more V-rings, plus male and female end rings, are fitted into an accurately machined gland giving the proper radial clearance with the piston rod or shaft. The chevrons are molded with a sharp edge which faces the pressure side. Only sufficient pressure to insure contact between this edge and the rod is supplied by the gland nut. Pressure from the fluid causes the packing to hug the rod or shaft, higher pressure causing a tighter seal. Here again, maladjustment of the gland can cause excessive friction. To prevent such maladjustment the packing is frequently spring-loaded, the depth of the gland being fixed, *Fig.* 11. While the V-ring is a great improvement over the stuffing box in reducing friction, there is still considerable drag at high pressures. Another disadvantage is that it does not seal well at low pressures unless considerably preloaded by spring or nut.

V-rings or chevron packings are molded from a variety of materials in a great many sizes. They are available from stock molds in sizes ranging from 1/8-inch inside diameter up to several feet. The width of the section, the "stack height", and also the shape vary with the ma-

Table III

Gland Dimensions for V-Ring and U-Cup Packings

(Dimensions Indicated on Fig. 15)

V-Ring or U-Cup				Cylinder or Rod Gland I.D. +.002 —.000 (in.)	Piston Rod or Piston Gland O.D. +.000 —.002 (in.)	Piston O.D. +.000 —.002 (in.)	Rod Bearings I.D. +.002 —.002 (in.)
	Nominal Size						
Dash No.*	W† (in.)	I.D. (in.)	O.D. (in.)				
1	3/16	1/8	1/2	.502	.123	.498	.127
2	3/16	3/16	9/16	.5645	.1855	.5605	.1895
3	3/16	1/4	5/8	.627	.248	.623	.252
4	3/16	5/16	11/16	.6895	.3105	.6855	.3145
5	3/16	3/8	3/4	.752	.373	.748	.377
6	3/16	7/16	13/16	.8145	.4355	.8105	.4395
7	3/16	1/2	7/8	.877	.498	.873	.502
8	1/4	1/4	3/4	.752	.248	.748	.252
9	1/4	5/16	13/16	.8145	.3105	.8105	.3145
10	1/4	3/8	7/8	.877	.373	.873	.377
11	1/4	7/16	15/16	.9395	.4355	.9355	.4395
12	1/4	1/2	1	1.002	.498	.998	.502
13	1/4	9/16	1 1/16	1.0645	.5605	1.0605	.5645
14	1/4	5/8	1 1/8	1.127	.623	1.123	.627
15	1/4	11/16	1 3/16	1.1895	.6855	1.1855	.6895
16	1/4	3/4	1 1/4	1.252	.748	1.248	.752
17	1/4	13/16	1 5/16	1.3145	.8105	1.3105	.8145
18	1/4	7/8	1 3/8	1.377	.873	1.373	.877
19	1/4	15/16	1 7/16	1.4395	.9355	1.4355	.9395
20	1/4	1	1 1/2	1.502	.998	1.498	1.002
21	1/4	1 1/16	1 9/16	1.5645	1.0605	1.5605	1.0645
22	1/4	1 1/8	1 5/8	1.627	1.123	1.623	1.127
23	1/4	1 3/16	1 11/16	1.6895	1.1855	1.6855	1.1895
24	1/4	1 1/4	1 3/4	1.752	1.248	1.748	1.252
25	5/16	1 1/4	1 7/8	1.877	1.248	1.873	1.252
26	5/16	1 3/8	2	2.002	1.373	1.998	1.377
27	5/16	1 1/2	2 1/8	2.127	1.498	2.123	1.502
28	5/16	1 5/8	2 1/4	2.252	1.623	2.248	1.627
29	5/16	1 3/4	2 3/8	2.377	1.748	2.373	1.752
30	5/16	1 7/8	2 1/2	2.502	1.873	2.498	1.877
31	5/16	2	2 5/8	2.627	1.998	2.623	2.002
32	5/16	2 1/8	2 3/4	2.752	2.123	2.748	2.127
33	5/16	2 1/4	2 7/8	2.877	2.248	2.873	2.252
34	5/16	2 3/8	3	3.002	2.373	2.998	2.377
35	5/16	2 1/2	3 1/8	3.127	2.498	3.123	2.502
36	3/8	2 1/2	3 1/4	3.252	2.498	3.243	2.502
37	3/8	2 5/8	3 3/8	3.378	2.622	3.372	2.628
38	3/8	2 3/4	3 1/2	3.503	2.747	3.497	2.753
39	3/8	2 7/8	3 5/8	3.628	2.872	3.622	2.878
40	3/8	3	3 3/4	3.753	2.997	3.747	3.003
41	3/8	3 1/8	3 7/8	3.878	3.122	3.872	3.128
42	3/8	3 1/4	4	4.003	3.247	3.997	3.255
43	3/8	3 3/8	4 1/8	4.128	3.372	4.122	3.378
44	3/8	3 1/2	4 1/4	4.253	3.497	4.247	3.503
45	3/8	3 5/8	4 3/8	4.378	3.622	4.372	3.628
46	3/8	3 3/4	4 1/2	4.503	3.747	4.497	3.753
47	3/8	3 7/8	4 5/8	4.628	3.872	4.622	3.878
48	7/16	3 7/8	4 3/4	4.753	3.872	4.747	3.878
49	7/16	4	4 7/8	4.878	3.997	4.872	4.003
50	7/16	4 1/4	5 1/8	5.128	4.247	5.122	4.253
51	7/16	4 1/2	5 3/8	5.378	4.497	5.372	4.503
52	7/16	4 3/4	5 5/8	5.628	4.747	5.622	4.753
53	7/16	5	5 7/8	5.878	4.997	5.872	5.003
54	7/16	5 1/4	6 1/8	6.128	5.247	6.122	5.253
55	7/16	5 1/2	6 3/8	6.378	5.497	6.372	5.503
56	1/2	5 1/2	6 1/2	6.503	5.497	6.497	5.503
57	1/2	5 3/4	6 3/4	6.753	5.747	6.747	5.753
58	1/2	6	7	7.003	5.997	6.997	6.003
59	1/2	6 1/4	7 1/4	7.253	6.247	7.247	6.253
60	1/2	6 1/2	7 1/2	7.503	6.497	7.497	6.503
61	1/2	6 3/4	7 3/4	7.753	6.747	7.747	6.753
62	1/2	7	8	8.003	6.997	7.997	7.003
63	1/2	7 1/4	8 1/4	8.253	7.247	8.247	7.253
64	1/2	7 1/2	8 1/2	8.503	7.497	8.497	7.503
65	1/2	7 3/4	8 3/4	8.753	7.747	8.747	7.753
66	1/2	8	9	9.003	7.997	8.997	8.003
67	1/2	8 1/2	9 1/2	9.503	8.497	9.497	8.503
68	1/2	9	10	10.003	9.997	9.997	9.003
69	1/2	9 1/2	10 1/2	10.503	9.497	10.497	9.503
70	1/2	10	11	11.003	9.997	10.997	10.003
71	1/2	10 1/2	11 1/2	11.503	10.497	11.497	10.503
72	1/2	11	12	12.003	10.997	11.997	11.003

*Army and Navy Specifications AN6225, AN6228, AN6229 for V-Rings and AN6226 for U-Cups.

W† denotes width of cross-section.

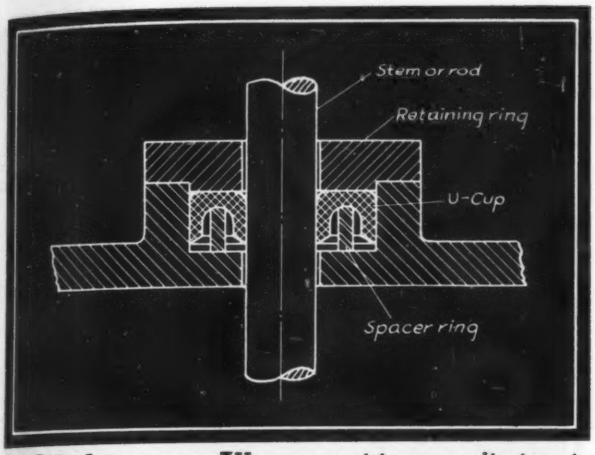


Fig. 13—Low pressure U-cup, requiring no adjustment

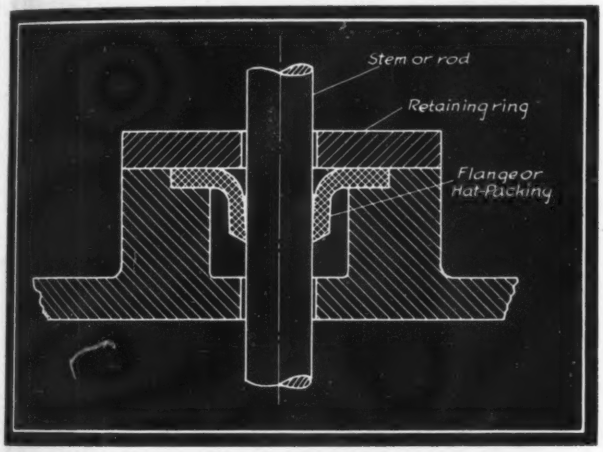


Fig. 14—Hat packing is a variation of U-cup, being one-half the latter in its shape

terial, size, and also the particular molding company making the packing. The Army and Navy have jointly standardized on V-rings for aircraft hydraulic systems. These standards are available as drawing number AN-6225.

Gland dimensions for V-rings molded to AN-6225 dimensions are given in TABLE III for the dimensions indicated in *Fig.* 15. This table also gives the nominal sizes of the V-rings listed in AN-6225. The length of the gland is determined by the number of rings required (stack height), the size of the male and female adapters, *Figs.* 10, 11 and 17, and the holding means employed for the packing.

A self-energizing packing, the V-ring tends to seal more tightly with increased pressure. It is necessary, however, to have enough initial pressure on the lips to prevent leakage before the pressure builds up. This is accomplished by preloading the stack of rings either with an adjustable nut or a spring load as in *Figs.* 10 and 11. Since it is not practical to mold the rings to close stack-height dimenisons, an adjustable gland length is necessary for the seal.

In large hydraulic press installations the V-rings usually are cut so that they may be assembled into the press without disassembling the piston and cylinder. The reader is referred to an article on V-rings, entitled "Design and Materials for Hydraulic Packings"[3].

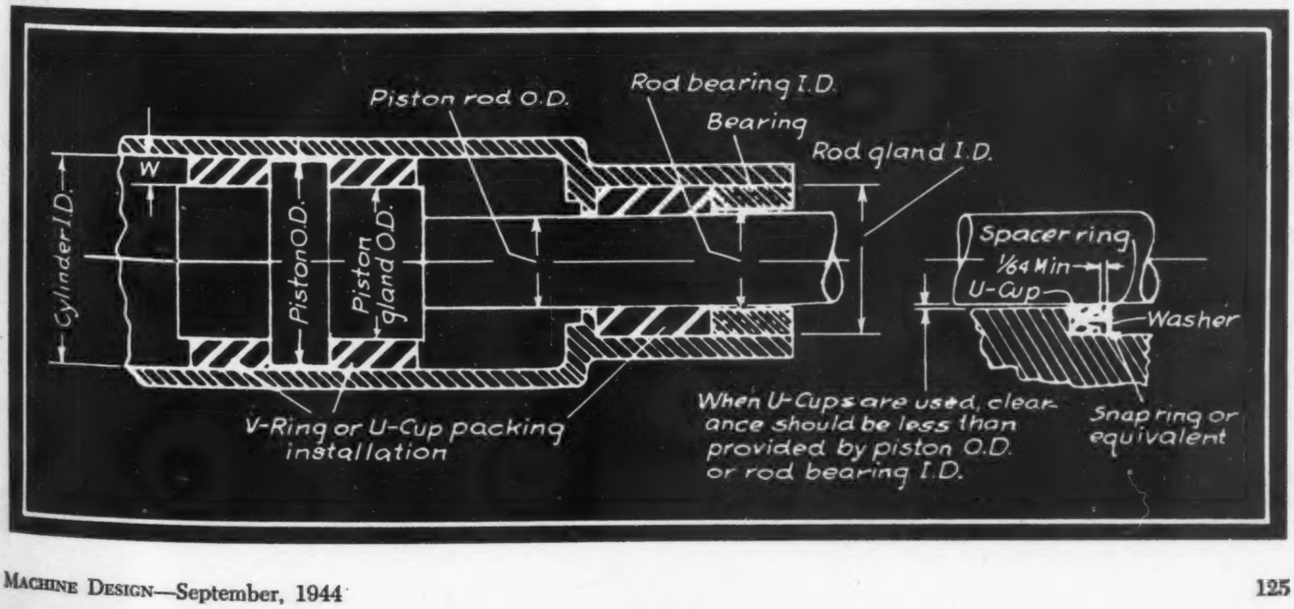
U-Cup Packings

In the medium and low-pressure field a packing that has found considerable favor is the U-cup packing, illustrated in *Fig.* 13. This is a molded packing in which the legs are made with an outward divergence so that when installed in the gland both legs will press against their respective sides of the gland. The ends of the legs are beveled to form relatively sharp edges to effect the seal against the rod and gland. A support ring of metal or plastic is used to hold the ends of the legs from the bottom of the gland. Gland length is fixed, no adjustment being necessary or possible with this type of packing.

The Army and Navy have standardized on U-cup packings in sizes up to 3-inch inside diameter for aircraft hydraulic systems. These packings are limited to 600 pounds per square inch in installations subject to approval by Army or Navy inspection. The standard drawing is AN-6226. Gland size is listed in TABLE III. U-cups are

[3] J. N. Smith—"Design and Materials for Hydraulic Packings", *Aero Digest*, April, 1942, Page 119.

Fig. 15—V-ring and U-cup gland designs for dimensions listed in TABLE III



produced also in larger sizes than the AN-6226 standards, and design dimensions for these are available from their manufacturers.

A variation of the U-cup is the flange or hat section packing shown in *Fig.* 14, being essentially one-half of a U-cup. The lip is molded at an angle to give a pressure against the rod. The packing is held by squeezing the flange by a retainer ring or cap.

O-Ring Packings

The so-called O-ring or "doughnut" packing has, in the last few years, become extremely popular with designers of aircraft hydraulic equipment. It is used widely as a piston-rod, piston and cylinder-head packing, as illustrated in *Figs.* 3, 4, 16, and 20. It has been used also as a packing in swing check and poppet check valves, and as a seal for separating internal elements in complex hydraulic valves. The chief advantage derived from the use of O-rings, particularly in dynamic seals, is simplicity of design. *Figs.* 9 and 11 compared with *Fig.* 16 illustrate this point with respect to packing and V-ring seals.

When properly installed in well designed grooves the packing friction of the O-ring is very low. Tests on available equipment have indicated a dynamic friction of 10 pounds per linear inch of packing for pistons ranging in size from 1⅛ inches to 2 inches in diameter. For a 2-inch cylinder this represents a total friction load of 62.824 pounds at a hydraulic pressure of 300 pounds per square inch.

As the pressure increases, this friction load will, of course, also increase. The static or "break-away" friction will be higher and will vary with the length of time the packing has been at rest. This "break-away" friction will also be dependent on the kind of material used in the ring.

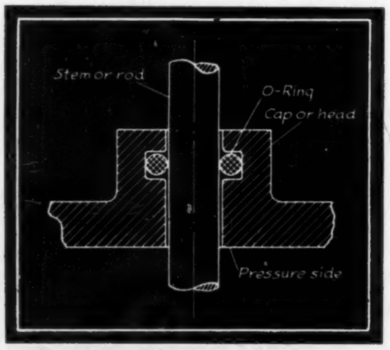


Fig. 16 — Simple, effective O-ring seal for piston rods requires no nut

The Army and Navy have jointly standardized on O-ring sizes for aircraft hydraulic equipment. These standards are included in AN-6227 and AN-6230, the former for dynamic seals and the latter for static seals. The gland sizes recommended for use with these packings are given in TABLE II for the dimensions shown in *Fig.* 12. Either the square or the V-groove can be used for pressures up to 1500 pounds per square inch. The V-groove is superior and gives longer life at higher pressures, but is somewhat more difficult to machine to accurate dimensions, particularly as a piston-rod gland.

The use of an O-ring as a piston-rod seal is shown in *Fig.* 16. This is the simplest of all rod packings and has proved very effective. Either a V-groove or a rectangular groove is used.

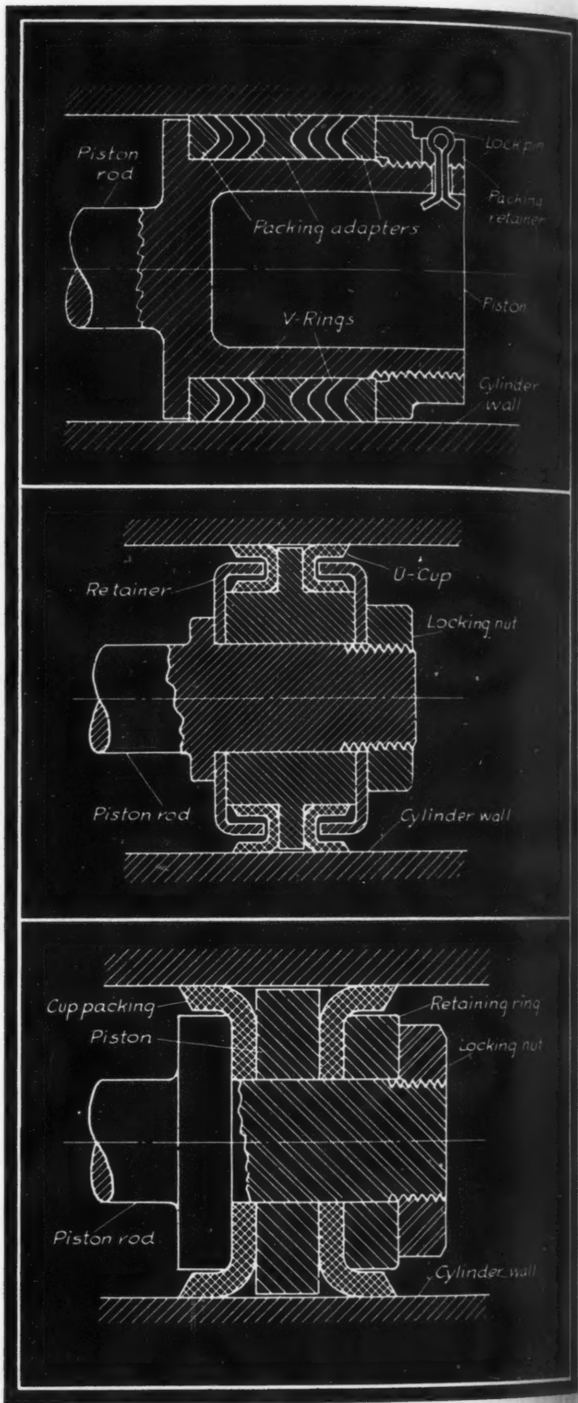


Fig. 17—Top—V-ring packing provides effective seal between piston and double-acting cylinder

Fig. 18—Middle—Double-acting U-cup seal for pistons

Fig. 19—Above—Cup packing assembly employed for sealing double-acting cylinder

In designing glands or grooves for O-rings, they should be so placed that assembly of the ring into the groove can be accomplished without damaging the ring surface. It is important that it be free from cuts or nicks to prolong life and assure proper seal.

To insure long life the surface against which the ring

slides should be smooth. To date the only surfaces that have worked well with O-rings as a dynamic seal are heat-treated steel, honed to a mirror finish, chrome plated, ground and polished. Copper, aluminum, and cast iron alloys have shown a tendency to scratch, and the packing life against these metals has been relatively short for applications made.

Piston Seals

With the exception of the flange or hat packing, all the dynamic seals discussed for piston rods are also used on pistons of double-acting cylinders, as illustrated in *Figs.* 17 to 20. *Fig.* 21 illustrates a packingless piston. The leakage across such a piston is a function of the clearance between piston and cylinder wall and the length of the piston. The use of annular grooves, as illustrated, minimizes the tendency of the piston to hug one side of the cylinder and therefore decreases friction, but this

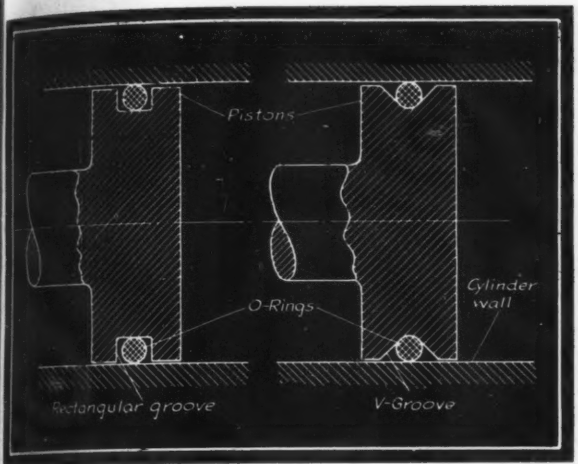
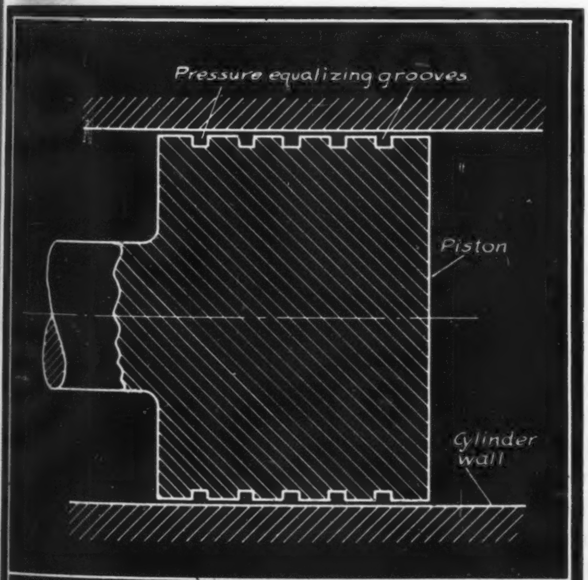


Fig. 20—Above—Single O-ring packings may be utilized with either rectangular or V-grooves

Fig. 21—Below—Packingless piston-design with pressure equalizing annular grooves

Fig. 22—Bottom—Automotive type piston ring is employed as a seal in this design



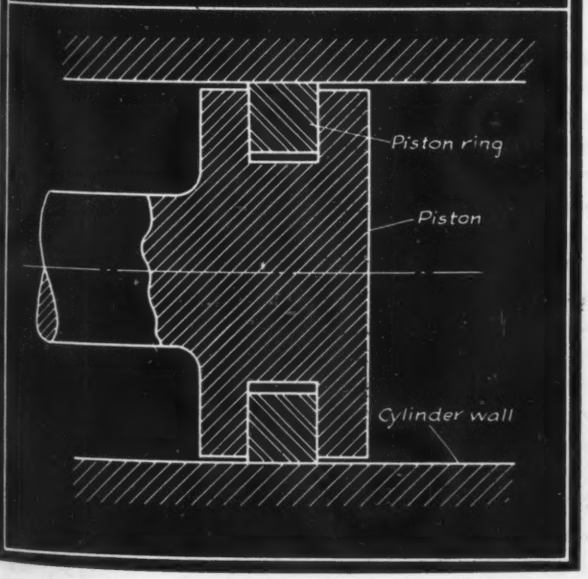
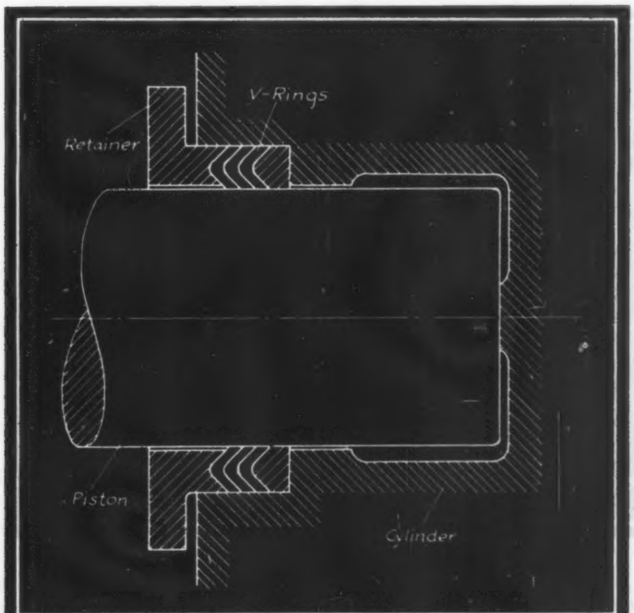
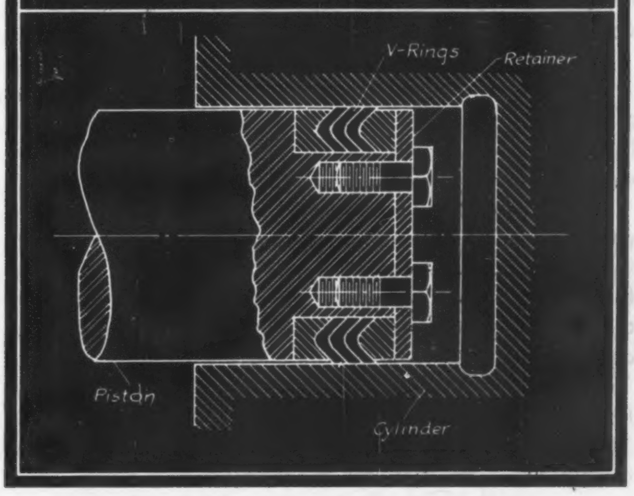


Fig. 23—Below—To facilitate service, packings for large single-acting cylinders are frequently designed like those for sealing employed piston rods

Fig. 24—Bottom—Inside-packing design of the type used for small single-acting cylinders





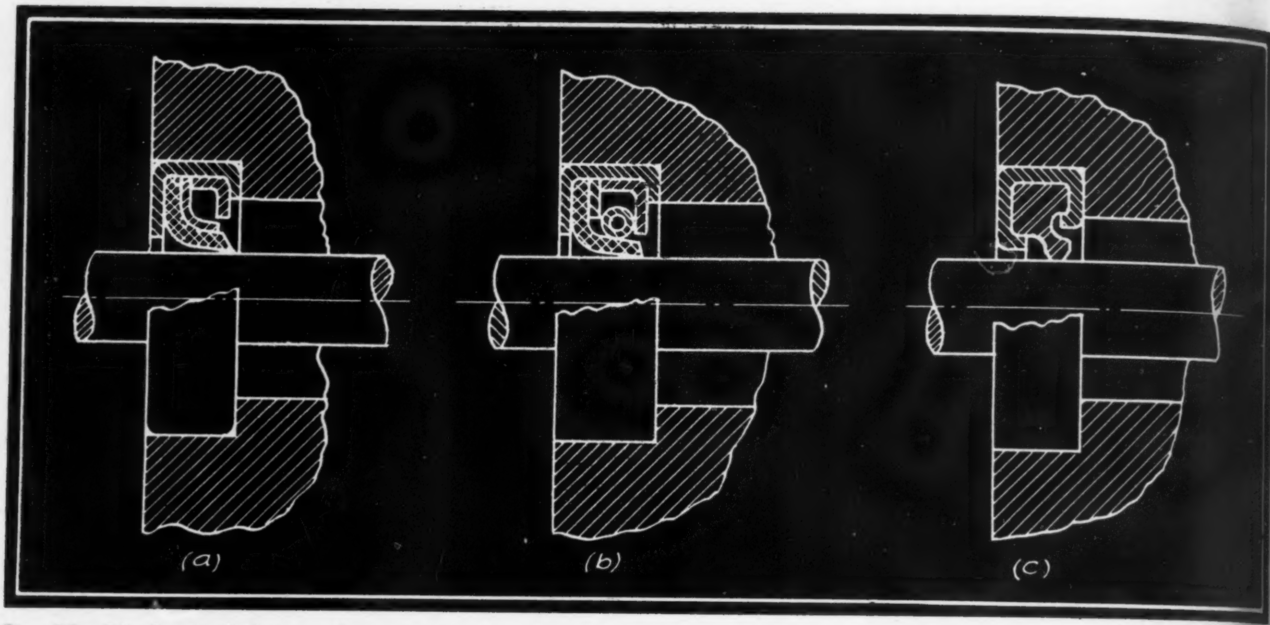
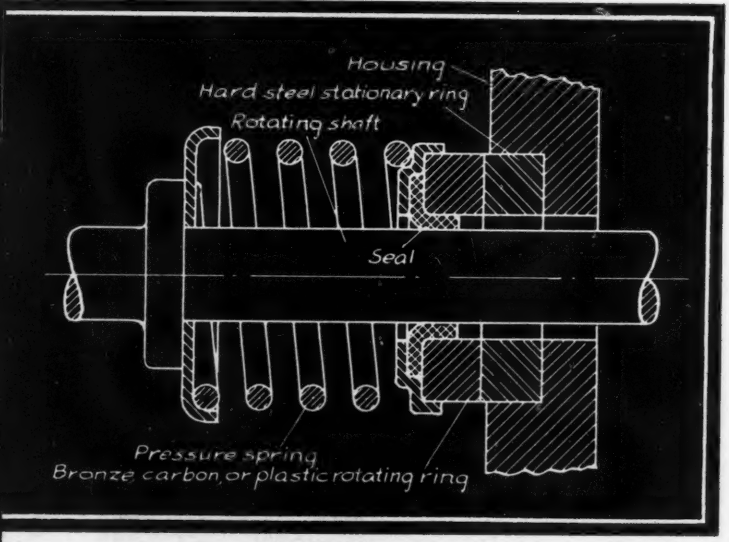


Fig. 25—Mechanical devices for causing lips of seal to hug shaft; at (a) is a leaf spring, (b) a garter spring and (c) a rubber spring

Fig. 26—Spring-loaded seal having rotating and stationary rings, obviating a rotating seal on the shaft



decrease in friction is obtained at the expense of increased leakage.

In addition, the cup packing is a widely used type. In small cylinders it is not necessary to fasten the cup to the piston. The oil pressure holds it in place. Automobile hydraulic brake cylinders are a typical example of a loose-cup design.

Pistons in double-acting cylinders are frequently packed or sealed by one or more automotive type piston rings, *Fig.* 22. These form effective seals if carefully fitted, but are of little value if the groove is wider than the ring and particularly if the joint in the ring is not accurately fitted.

In large single-acting cylinders the piston packing is frequently designed like a piston-rod packing, *Fig.* 23. This makes it possible to service these installations more easily. Small units do not benefit by such a design from a service point of view and are therefore more generally designed with "inside" packing, *Fig.* 24. In small units machining and assembly operations are simplified with inside packing.

Rotating Shaft Seals

So far, only the transverse type of packing or seal has been discussed. There are many designs of packings or seals for rotating or oscillating shafts. Frequently the piston-rod packings illustrated in *Figs.* 8, 9, 10, 11, 13, and 14 can be used with rotating shafts. For sealing at high hydraulic pressures the V-ring has been found most effective.

Where the shaft seal is primarily to prevent the loss of fluid under pressures not exceeding 5 pounds per square inch, standard shaft seals are most satisfactory. All of those using plastic or organic packing materials seem to be variations of the flange or hat packing, with various mechanical devices such as garter springs, leaf springs, or rubber springs to cause the lip or edge of the packing to hug the shafts.

The detail shape of the packing lip, the type and hardness of the material, and the amount of spring pressure has a great bearing on the amount of heat generated for a given installation. Surface speeds in this type of packing may be high; several thousand feet per minute are frequently encountered. Several cross sections of these packings are illustrated in *Fig.* 25, showing leaf spring at *a*, garter spring at *b* and rubber spring at *c*.

Where low friction, high speeds, and high suction heads are present, spring-loaded metallic, carbon, or hard plastic seals are finding considerable favor. A typical design is illustrated in *Fig.* 26. The stationary casing carries a hard, smooth, circular plane surface against which rotates a ring of low-friction metal, carbon, or plastic. This metal ring is pressed against the casing surface by a spring and is keyed or otherwise fastened to the shaft. It is also sealed to the shaft by either a rubber bushing or a flexible tube. This type of seal is frequently used on the drive shaft of hydraulic pumps.

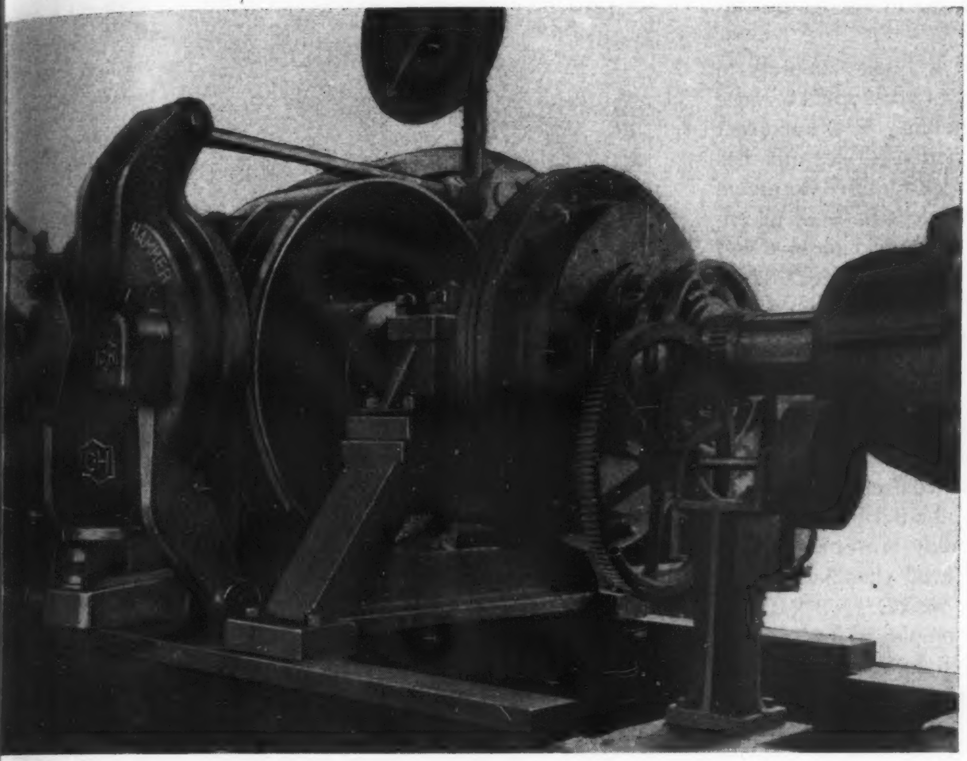


Fig. 1—Application of a magnetic brake to a coal-yard hoist

Specifying Intermediate Components for Machine Drives

By Richard K. Lotz

Part II—Brakes

MANY of the principles embodied in the design and functioning of brakes are similar or identical to those employed in the development and utilization of clutches. Because friction brakes, like friction clutches, depend for their action upon friction between surfaces of driving and driven members, their capacities are predicated on unit pressures developed between the friction surfaces, the coefficient of friction possessed by the friction surfaces and the ability of the unit to dissipiate heat. This last mentioned factor is of considerable importance because, if a brake of insufficient capacity is used, heat caused by friction on the sliding surfaces may well grow to a magnitude sufficient to burn out the brake lining material and thus cause an expensive shutdown of equipment.

Design data and calculations applicable to the basic forms of block, disk and band brakes are dealt with so thoroughly in existing literature that no attempt will be made here to cover this familiar ground. Rather, it is intended to present the principles of operation together with

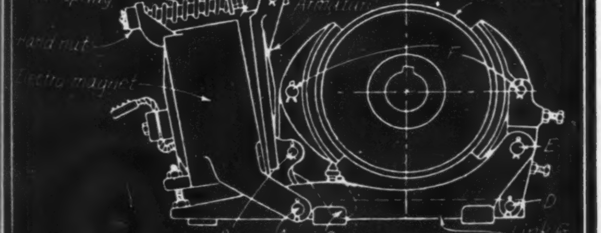


Fig. 2—Power of this magnetic brake is applied by spring when the magnet is de-energized. Link G equalizes shoe pressures on brake wheel

application information applying to the more outstanding types of brakes.

Generally, the designer of a brake attempts to put the required braking power into as small a space as good design practice will permit. It is important to recognize that size of the friction surface area has no theoretical effect on the force required to create a given amount of friction but *does* have an all-important effect on the heating of the friction area and the rate at which that area can be cooled. Thus, the brake designer must balance his design between compactness on the one hand and ample cooling area on the other.

Two Predominant Types

Just as there are radial and axial clutches, so are there radial and axial brakes. Radial brakes can be internal or external and generally are either of the block or band type. In the axial classification are found the cone and disk-type brakes.

Many different means are employed to energize brakes. There are, for example, electromagnets, electric motors, hydraulics, pneumatics, and mechanical means (such as toggles, links and cams). The decision as to what type of power will be used to energize a brake for a given application will depend not only on the characteristics peculiar to the various types, but on what type of power is conveniently available and what kind of a braking job the unit will be called upon to do. Thus, in the case of magnetic brakes, a prime influencing factor might be the necessity for a type of brake unit which automatically will bring all equipment to a quick stop when there is an electric power failure. In other words it often is looked upon as a safety device. Such brake units are pictured in *Figs.* 1, 2 and 4.

Referring to *Fig.* 2, to establish brake release, the electromagnet is energized and the armature, as it is drawn to the magnet, pivots at bearing *A*, pulling the left-side shoe lever to the left by means of pin *B*, pulling the left shoe away from the brake wheel and pulling link *G*. Link *G* pulls the lower end of the right-side shoe lever (pivoting on bearing *E*) to the left, thus moving the right-hand brake shoe away from the brake wheel.

Energization of this brake is accomplished by cutting off the electric current to the magnet. The power spring then comes into play, pushing the armature away from the magnet and, through the linkage already described, forcing both of the brake shoes against the brake wheel with equal force. This particular design has good power capacity due in great measure to the fact that half of the brake-wheel surface is always freely exposed to the atmosphere for heat radiation. All of the requisite adjustments are readily accessible and the brake wheel can be removed from between the shoes either by turning on the electromagnet or by tightening up the handnut. An application of seven such brakes is shown in *Fig.* 3.

Electromagnet coils for magnetic brakes are shunt wound for connection across the line voltage or for connection in series with the main motor armature. Shunt windings are made for continuous duty or for intermittent (1 hour) duty. Brake torque rating varies according to

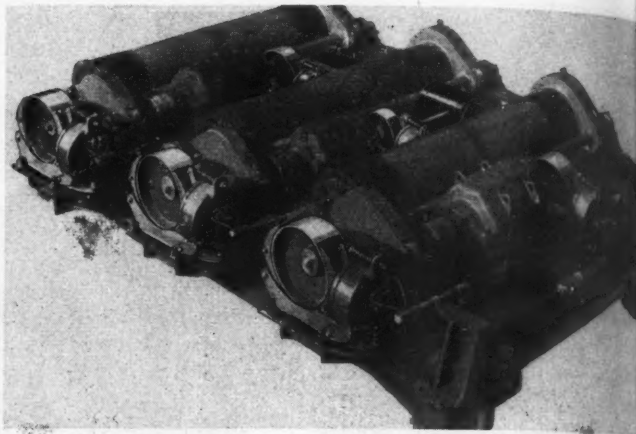
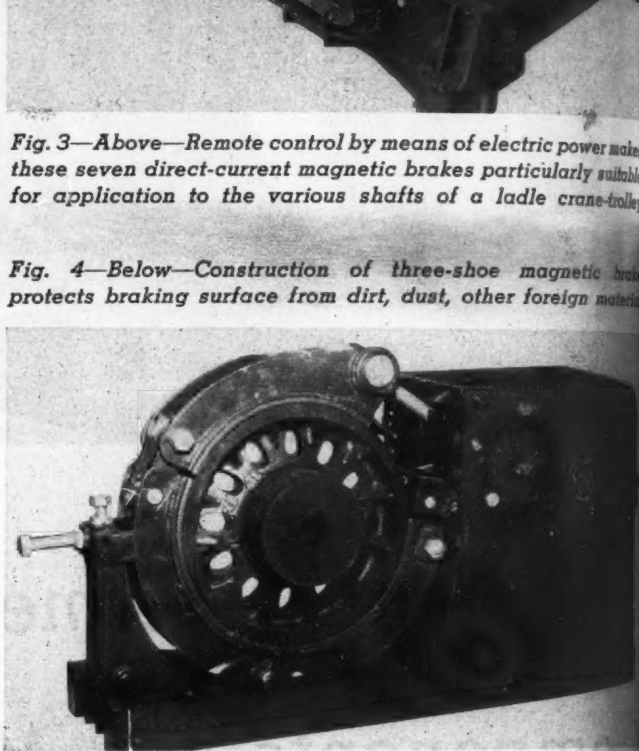


Fig. 3—Above—Remote control by means of electric power makes these seven direct-current magnetic brakes particularly suitable for application to the various shafts of a ladle crane-trolley

Fig. 4—Below—Construction of three-shoe magnetic brake protects braking surface from dirt, dust, other foreign material



magnet capacity which depends on the duty cycle. Series windings are made for 1-hour duty, for half-hour duty, or for unique duty cycles as calculated from the current-vs-time chart for the installation.

Magnetic brake selection has been standardized generally so that where a given motor frame is used, on the usual type of installation, a corresponding brake is employed. This simplifies the selection problem for most jobs to the reading of a table. For unique applications, general points to be considered in the selection of a brake are the torque and the heat-dissipating balance between the particular requirements and the capacities of available brakes.

It is necessary only infrequently to use a brake with a higher torque rating than the torque calculated from the horsepower of the motor and the speed of the shaft on which the brake is to be mounted. This is true in spite of the fact that the ultimate torque of the motor is several times the rated value, because the motor is selected with a liberal safety factor and because the mechanical inefficiency works against the motor and helps the brake. Thus, for a mechanical efficiency of 70 per cent the requirement for a brake is less than 49 per cent of what the motor must deliver.

In machines having high inertia loads, the driving motor must not only keep the machine rolling but must accel-

erate it from a standstill to operating speed. Thus, when the brake is selected on the basis of motor size it has some extra energy-absorbing capacity above that which would exist if selection were based on the mechanical load. This compensating factor is increased still further if dynamic braking control is provided, since the motor in such a case must be still larger to get rid of the extra heat developed during dynamic braking. The brake corresponding to the larger motor is again larger and has less to do because dynamic braking takes over a large share of the job of deceleration. Under adverse conditions without dynamic control, with normal machine friction, and with half of the cycle time spent accelerating and half decelerating, a brake needs 28 square inches of friction surface per horsepower of motor rating. Standard commercial brakes are designed primarily as holding brakes and have from 2 to 3 square inches per horsepower. Because of the complexity of considering the kinetic energy factor, it quite often is ignored and faulty operation occurs on the occasional installation which delivers far more energy to the brake than it can dispose of at reasonable temperatures.

In exacting conditions, the time of action of the brake is to be kept in mind. Some part of a second elapses after the brake coil circuit is closed before the brake releases. This varies with the size of brake, the type of winding and the adjustment. Similarly some time elapses after opening the coil circuit before the brake grips. Timing requirements should be checked carefully to assure a satisfactory installation.

In every brake structure it is necessary to make adjustments to compensate for wear of friction faces. In a magnetic brake, as the lining wears away the magnet stroke increases and the brake becomes progressively slower. If the need for adjustment is ignored too long the magnet will fail to free the brake, the friction faces will overheat, and permanent damage may result. This is a maintenance question and proper attention can be encouraged if maximum accessibility and convenience are provided.

A unique but simple and effective means of brake energization is utilized in the internal single-band type brake illustrated in *Fig.* 5. The rotating drum part of this unit (not shown in the drawing), fits over the outside of the brake band with sufficient clearance existing between the inside diameter of the drum and the band to permit free rotation.

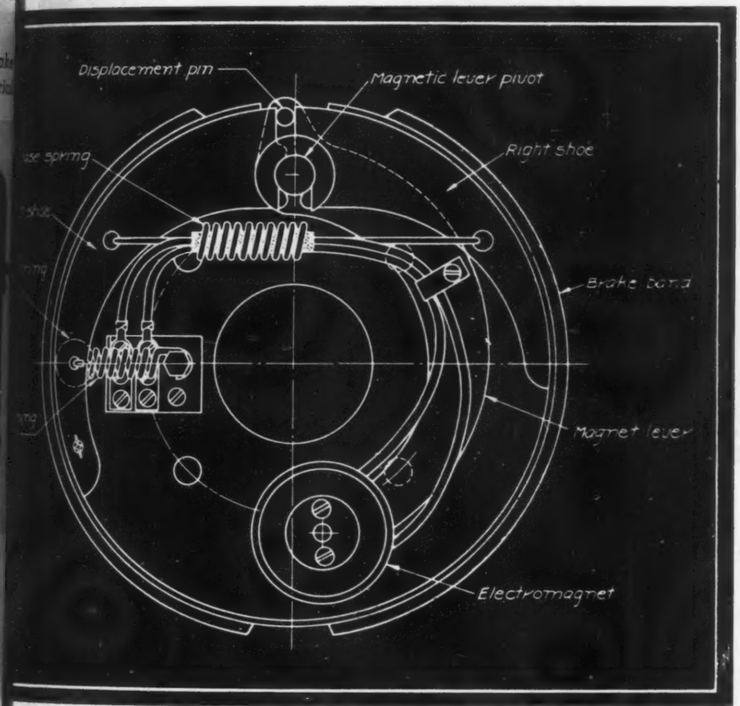
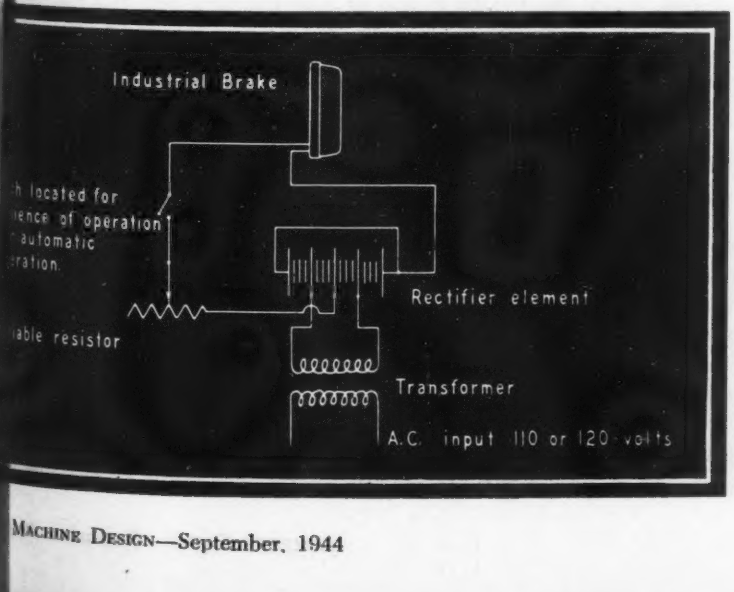


Fig. 5—Above—In this single-band electric brake, force exerted by electromagnet is amplified by means of leverage to displacement pin

Fig. 6—Below—Typical wiring diagram for electric brake above shows how alternating current is rectified and controlled by rheostat



Employs Magnet on Lever

Braking action is effected through the electromagnet, its lever and pin arrangement, and the shoes which are fastened directly to the brake band. On the inner face of the rotating-drum part of the brake is affixed an armature in the form of a flat, smooth, circular plate. The electromagnet, when not energized, slides lightly along the surface of the armature and no braking action takes place. However, when electric current is fed through the coil of the magnet, the magnetic attraction between the magnet and the armature establishes a definite pressure between these units and the resulting friction carries the magnet along in the direction of drum rotation. Thus, in the unit shown in *Fig.* 5, the magnet and the lever to which it is attached would move to the left, swinging about the magnet lever pivot and urging the displacement pin to the right. It will be seen that the right and left shoes are separate pieces, each held against the magnet lever pivot by the release spring. Since the displacement pin is located between the shoe ends, when it moves to the right it will force the right shoe away from the pivot. This serves to swing the entire unit clockwise around the pivot until the brake-band lining at the left side touches the drum (an extreme-

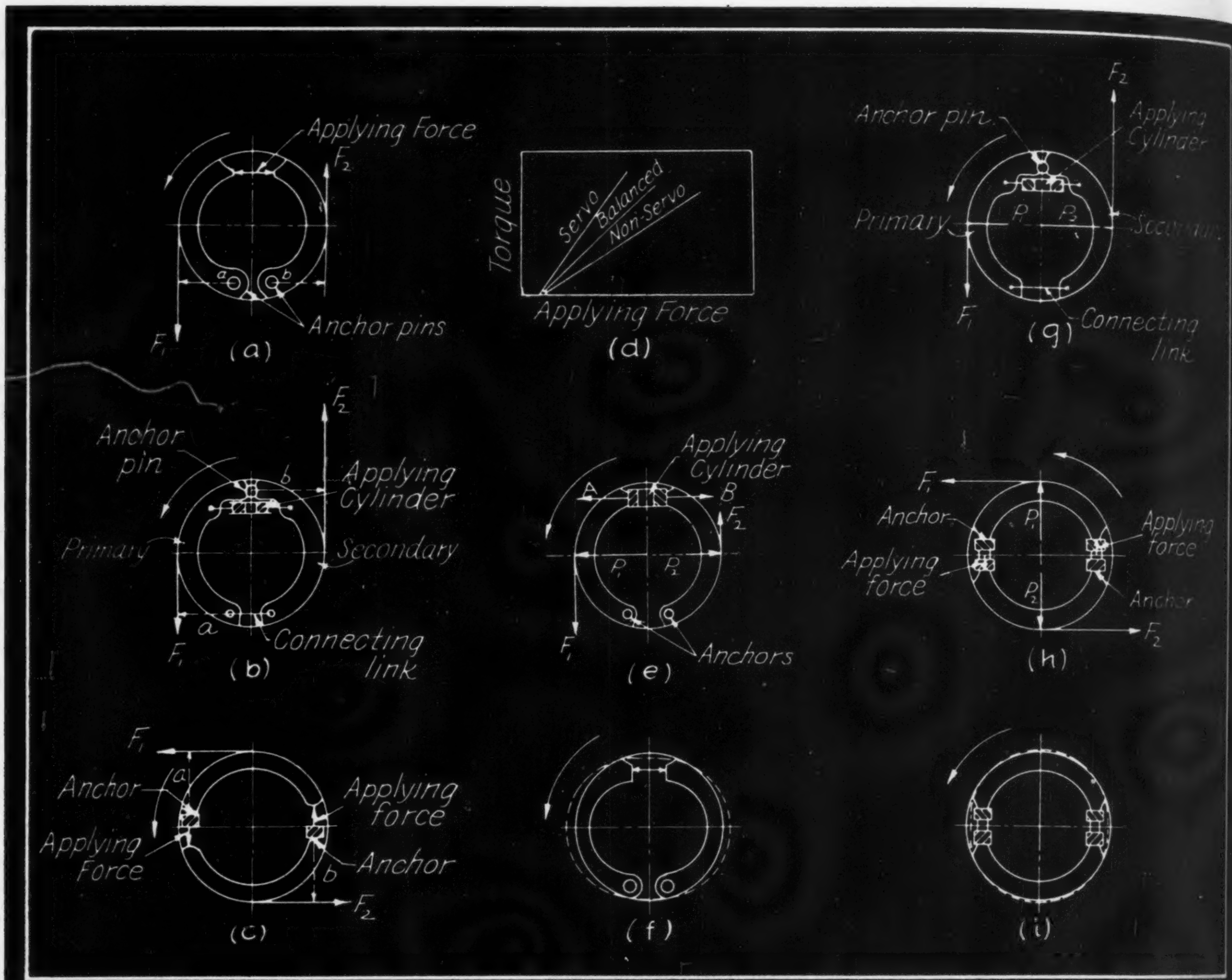


Fig. 7—Diagrams show how servo, nonservo and balanced brakes of the two-shoe type work. Distortion to which drums of the different brakes are subject also is indicated

ly small movement), and spreads the brake band, forcing it against the brake drum and thus creating the braking friction.

Because of the direction of rotation of the drum, the friction between it and the brake band will tend to make the band spread still further, thus creating braking power additional to that which results solely from the action of the electromagnet and its lever. This, of course, is a form of self-energization.

This brake is constructed of a minimum of simple parts, is extremely sturdy and offers a great deal of braking power for its size. One of its chief advantages lies in the fact that it can be rheostat-controlled from a remote source. The simplicity of the electrical circuit required in using such a brake is demonstrated by the typical wiring diagram shown in *Fig.* 6.

Brake Can Be Used as Clutch

It is interesting to note that by adding a collector ring on the shaft, thus permitting current to be drawn while the whole unit is rotating, this brake may be used as a powerful, slow-speed clutch. However, it is not recommended for applications where a constant slipping torque is required because the coefficient of friction of the brake lining changes as the lining heats up and, therefore, a constant torque cannot be maintained without constantly varying the current.

A brake which lends itself to a great variety of applications is the internal-shoe brake of which there are three principal types: Nonservo, servo and balanced. Each may be operated hydraulically or mechanically and possesses certain characteristics which make it particularly suitable for specific kinds of service.

Shown in *Fig.* 7(a) is a two-shoe, nonservo brake. The shoes are anchored at the bottom ends and spread apart at the top by means of a mechanical cam or a hydraulic wheel cylinder to press the shoes against the drum. When this is done while the drum is rotating counterclockwise as shown in the diagram, there is a frictional force produced at the drum, represented by F_1. Note that this force has a moment arm a about the anchor, which tends to rotate the left-hand shoe counterclockwise about its anchor, and thus press it harder into the drum. This self-energization results in an increase in the frictional force F_1, the same as if the applying force had been increased.

Applying the right-hand shoe produces a frictional force F_2 at the drum in the direction shown by the arrow. This force has a moment arm b about the anchor pin and tends to rotate the shoe counterclockwise and away from the drum. This shoe is termed "nonenergizing" and its action results in a decrease in the frictional force F_2 the same as

if the applying force had been decreased. The frictional force F_1 on the forward shoe may be several times greater than the frictional force F_2 of the reverse shoe, depending on the coefficient of friction of the linings of the two shoes. Thus a brake of this type contains a self-energizing shoe and a nonenergizing shoe, but no servo action.

Diagram (b) in *Fig.* 7 shows a servo-brake. In this type the shoes have a common anchor at their top ends. They are applied at a point near the anchor, and are connected at the bottom by a floating link which usually is made adjustable to compensate for lining wear. When the shoes are spread apart by the cam or hydraulic cylinder, and the drum is rotating in the direction shown by the arrow, a frictional force F_1 is produced at the drum. Since this shoe, which is known as the "primary" shoe, is not anchored to the backing plate at its lower end, it starts to rotate with the drum. As may be seen, this movement and force is transmitted to the other shoe, known as the "secondary" shoe, through the connecting link to become the applying force of the secondary shoe, providing servo action.

Servo-Brake Shoes Are Self-Energizing

It should be remembered that, although the primary shoe is not anchored directly to the backing plate, the frictional force F_1 produced by this shoe still acts as a retarding force on the drum and this force is added to that of the secondary shoe. Force F_1 has a moment arm a about the connecting pin between the shoe and link, tending to press it harder into the drum; therefore it is an energizing shoe. Force F_2 of the secondary shoe has a moment arm b about the anchor, tending to press it harder into the drum and is, therefore, also an energizing shoe. If the two shoes are lined with the same friction material, the frictional force F_2 of the secondary shoe will be about double that of the primary shoe F_1. In the opposite direction of braking, the secondary shoe becomes a primary shoe.

Shown in diagram (c) of *Fig.* 7 is a "balanced" brake, with the two shoes individually applied and individually anchored at diametrically opposite points. When the drum is rotating in the direction shown by the arrow, and the shoes are individually actuated to press them against the drum, the frictional force F_1 of the top shoe has a moment arm a about its anchor tending to press it harder into the drum; and the frictional force F_2 of the bottom shoe has a moment arm b about its anchor tending to press it harder into the drum; thus both shoes are self-energizing. However, this brake, like that shown at (a), develops no servo action.

If the applying forces and the types of linings on the two shoes are alike, the frictional forces F_1 and F_2 will be the same and both shoes will do the same amount of work. Also, if the brake is constructed with geometrically similar applying forces and anchoring points for the opposite direction of drum rotation, the brake will have the same effectiveness in both directions.

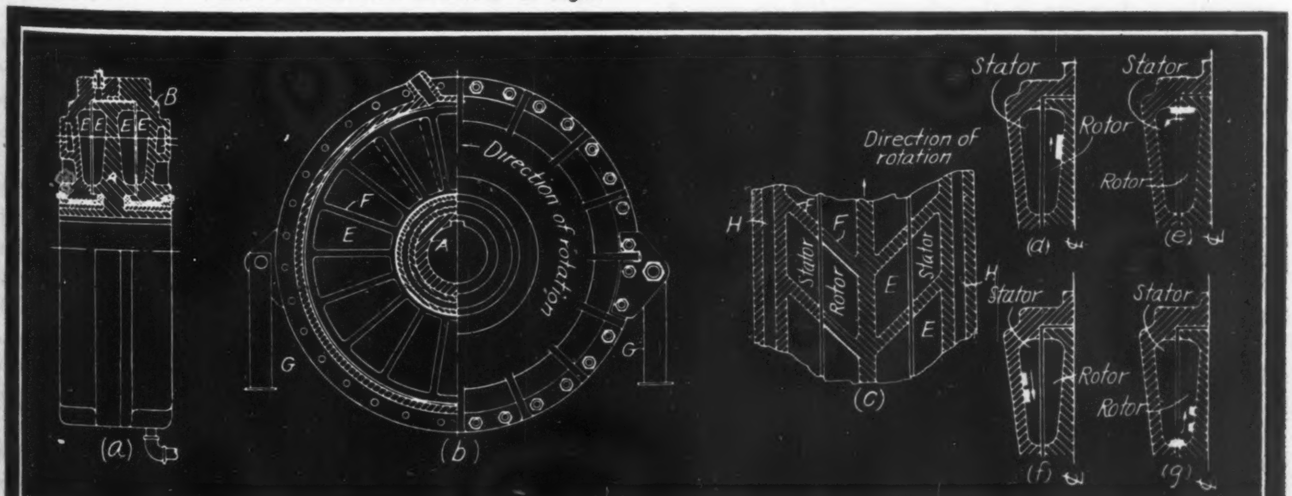
As a comparison of the torque capacity of the various types of internal-shoe brakes, a curve has been plotted as shown at (d) of *Fig.* 7. It shows that the balanced brake is somewhat more effective than the nonservo, and that the servo is more effective than the balanced brake. This can be attributed to the fact that the balanced brake has two self-energizing shoes against one self-energizing and one nonenergizing shoe of the nonservo brake, while the servo brake of course has servo action in addition to its two self-energizing shoes.

Brakes operating at high speeds require good controllability which exists when the input is proportional to the output over the entire torque-capacity range of the brake. This characteristic is inherent in the balanced brake, which therefore is the most suitable for this type of service.

Servo-Brake Suited to Low Speeds

In low-speed operation, controllability is not so important. Since the servo-brake is the most effective of the three types it is best adapted for this kind of service because it allows the use of the smallest diameter brake for a given torque. The nonservo brake, which is the least effective, is suitable in those applications where the torque requirements are not high and where unequal drum distortion and unequal lining wear between the two shoes

Fig. 8—Principal parts of a hydrodynamic brake are the stators and rotor. How "shearing" action on fluid develops retarding power is shown in the four sections at right



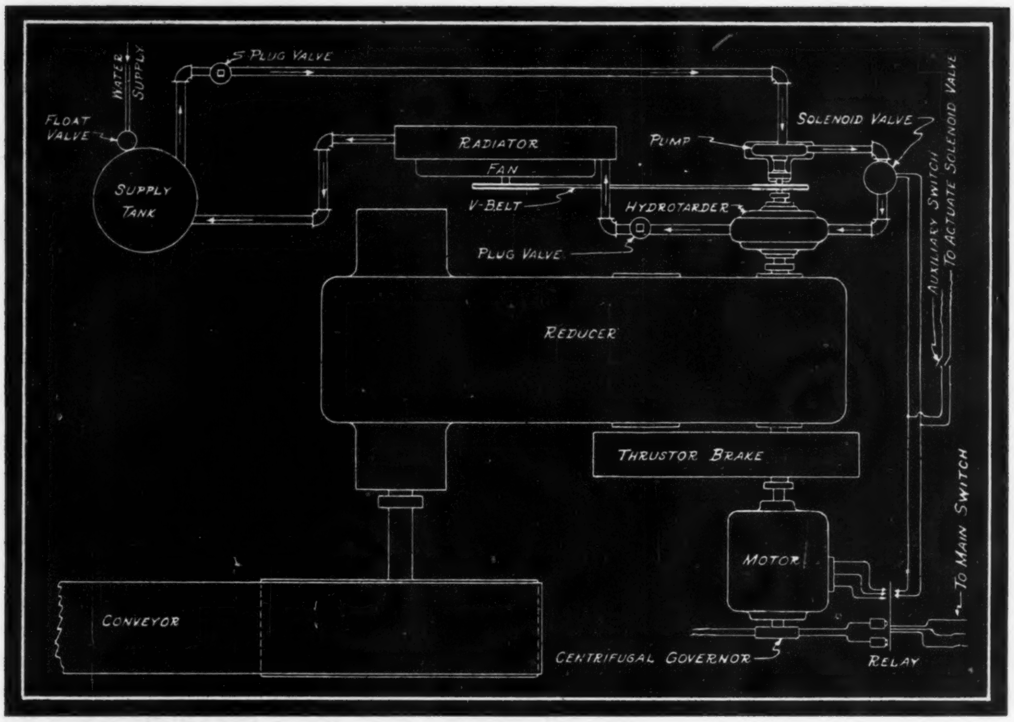


Fig. 9—Diagram illustrates how hydrodynamic brake can be used to limit speed of inclined belt conveyor

is not of great importance.

Since neither drums, shoes, anchors, nor backing plates can be made perfectly rigid, the forces imposed upon these various elements during a brake application cause deflections or distortions. It is assumed that the two shoes of the brake shown in *Fig.* 7 (e) are lined with materials having the same coefficient of friction and that a straight-bore wheel cylinder is used, that is, a wheel cylinder having the same diameter bore at each end. For a given line pressure, this straight-bore cylinder produces equal applying forces A and B on the two shoes. Statically, that is, with the drum not rotating, these applying forces produce equal radial forces, P_1 and P_2, on the two shoes. However, when the drum is rotating in the direction shown by the arrow, the frictional force F_1 of the forward shoe is increased due to its energizing action. Since this has the same effect as increasing the applying force a, it also has the same effect as increasing the radial pressure P_1. Conversely, the frictional force F_2 of the reverse shoe is decreased due to the nonenergization of this shoe, with a proportional decrease in the effective radial pressure P_2.

Unequal Distortion Means Loaded Bearings

Forces P_1 and P_2 cause the drum to distort outwardly at the points where these forces are greatest. Since force P_1 is greater than force P_2, the left side of the drum will be distorted outwardly farther than the right side. The distortion diagram is shown exaggerated by the dotted lines in *Fig.* 7 (f), the solid line representing the undistorted drum. The difference between the radial forces P_1 and P_2 constitutes a resultant force acting in the direction of the greater, or to the left in this case. This force is imposed upon the bearings. Since lining wear is a function of the radial pressure, the forward shoe, having the greater radial pressure, will wear faster than the reverse shoe. If both shoes of the servo-brake shown in *Fig.* 7 (g) are lined with materials having the same coefficient of friction, the friction force F_2 of the secondary shoe would be about twice the friction force F_1 of the primary shoe. Since here again the forces F_1 and F_2 are different, the radial pressures P_1 and P_2 also are different, with the result that the drum distortion diagram is similar to that shown at f for the nonservo brake, except that in this case the maximum distortion is on the right instead of the left side. The resultant force to the right imposes a load on the bearing and also causes the secondary shoe to wear faster than the primary shoe.

Balanced Brake Characteristics

In the balanced brake, *Fig.* 7 (h), if both shoes are lined with materials having the same coefficient of friction, and if the wheel cylinders or the applying means produce the same applying pressure at each shoe, then the frictional forces F_1 and F_2 of the respective shoes will be equal. Since both shoes have the same amount of self-energization, the radial pressures P_1 and P_2 also will be equal; therefore the lining wear of both shoes is the same. Since now the radial forces P_1 and P_2 are equal and opposite, the drum distorts symmetrically, as shown exaggerated by the dotted line in *Fig.* 7 (i), the solid line again being the undistorted drum. Thus this brake does not impose any load on the bearings.

One of the most recent developments in the brake field is the unit pictured cross-sectionally in *Fig.* 8. It is different from most other types of brakes in that it absorbs power by fluid friction rather than mechanical friction. Known as a "Hydrotarder" or "Hydromatic Brake", this hydrodynamic unit does not take the place of any of the regular mechanical brakes but supplements them where retarding action is required without actual stopping over a considerable period of time.

As shown in view a of *Fig.* 8, the unit consists of three main parts: Right and left-hand stators B and the rotor

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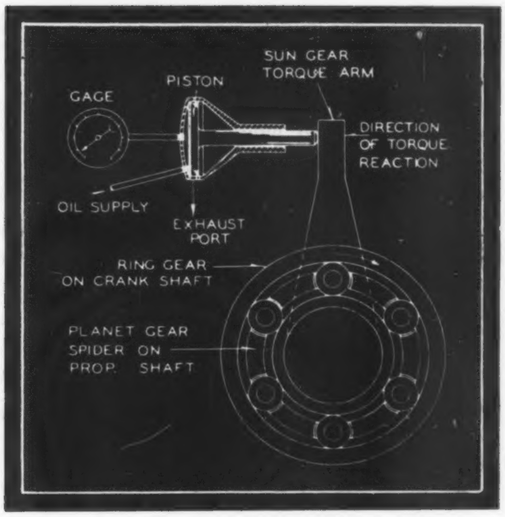


Fig. 1 — Hydraulic torquemeter indicates torque reaction on sun gear of a planetary transmission

Torquemeters Furnish Check on Machine Performance

By F. W. Godsey and B. F. Langer
Westinghouse Electric and Mfg. Co.

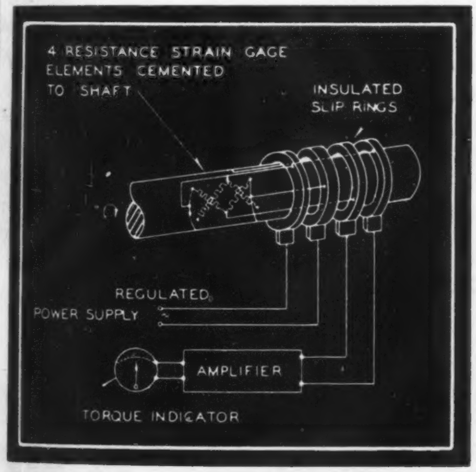


Fig. 2—Electrical torquemeter shown uses resistance-wire strain gages attached to the shaft whose twist it measures

TORQUE MEASUREMENTS may be used effectively in controlling many types of machines and processes, but real or fancied difficulties in applying torquemeters have stood in the way of their more widespread use. The accompanying article, an abstract of a recent A.I.E.E. paper, presents a brief discussion of the better-known types of aircraft engine torquemeters and introduces a new design of magnetic coupled torquemeter that recently has come through its development stage

TWO general types of torquemeters have been applied to aircraft engines to determine engine output; these are the hydraulic and the electrical types. The hydraulic torquemeter (1)*, (2), is the better known of the two and has been in use for a number of years; it usually measures the torque reaction on a stationary member of the propeller shaft reduction gear train by means of a hydraulic piston and a pressure-regulated oil supply. The oil pressure is controlled by means of slight piston movements to balance exactly the torque reaction of the selected gear, usually the stationary sun gear of a planetary gear set, *Fig.* 1, and the oil pressure which is proportional to the torque is measured by a conventional pressure gage. When double reduction gears are required or when operation at an extremely low temperature is necessary, and when excessive torsional vibrations are present, the hydraulic system becomes difficult to handle. Electrical torquemeters fall into three general classifications, the strain-resistive, strain-capacitive, and strain-magnetic.

The strain-resistive type may measure either the twist in the propeller shaft by means of strain-sensitive resistance elements mounted directly on the shaft or it can measure the deflections of a stationary reaction member, somewhat similar to the conventional hydraulic torquemeter in application except that the actual deflection of a spring is measured rather than the balancing oil pressure. If the strain-sensitive resistance elements are attached to a stationary reaction member, direct electrical connections can be made between the resistance units and the indicating instrument; but if the resistance units are mounted on a rotating shaft so as to be sensitive to the actual twist in a specified gage length of shaft, electrical connections must usually be made through slip rings on the shaft with brushes riding on the slip rings, *Fig.* 2.

Wire resistance strain gages (3) have the required accuracy,

*References in parentheses are listed at end of article.

but the resistance changes, for strains corresponding to maximum allowable stresses in both the engine parts and the wire resistance elements, seldom exceed one per cent of the initial resistance. Therefore, use of a sensitive potentiometer and balancing bridge is required, or high-gain electrical amplifiers must be used with direct-indicating instruments. The ease of application of the small resistance-wire strain gage and its general reliability make it valuable for laboratory work, but its low sensitivity appears to rule it out as an aircraft flight instrument. A further disadvantage is the relatively large temperature coefficient of resistance of most of the wire-resistance strain gages; normal temperature excursions in aircraft engines result in temperature-resistance changes of greater magnitude than the strain-resistance changes under measurement; these temperature effects can be eliminated, however, by careful adjustment of compensators.

Applying Strain-Capacitive Elements

Strain-capacitive elements are used in much the same manner as strain-resistive elements, and may be mounted either on rotating shafts with slip-ring connections or on torque-responsive stationary members of the engine. Here an air dielectric condenser is formed of two groups of plates or conducting surfaces insulated from each other and so positioned that when the assembly is subjected to strain, either the dielectric spacing between plates of opposite polarity changes or opposed plate areas alter in proportion to the strain, with a resultant change in the capacitance of the condenser. The per cent change in capacitance which can be obtained for permissible strains in engine parts can be large, and for that reason much effort has been expended in attempts to utilize strain-capacitive elements in strain gages and torquemeters. However, space and weight limitations usually restrict the capacitance of the strain sensitive element to a few micro-microfarads; and therefore even with alternating-current power supplies of several hundred kilocycles frequency, energy levels in the sensitive circuit elements are so low as to require amplifiers ahead of the torque-indicating instrument, with the usually attendant difficulties of maintaining calibration. Capacity-sensitive strain elements also must be hermetically sealed to exclude oil, moisture and dirt; and the uncertain stray capacities of electrical connections and of supports are major difficulties.

Strain-magnetic elements lend themselves to the measurement of small displacements much more readily than do any of the previously mentioned strain-sensitive devices when it is desirable to connect direct-indicating instruments to the strain-sensitive circuit without interposing amplifiers. The conventional magnetic strain gage operates at sufficiently high energy levels that its output can be utilized to drive an indicating instrument directly; in such manner it is commonly used as the sensitive element

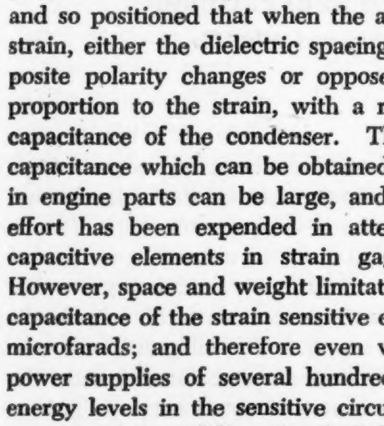


Fig. 3 — Simplified magnetic strain gage circuit is responsive to small movements of the armature A

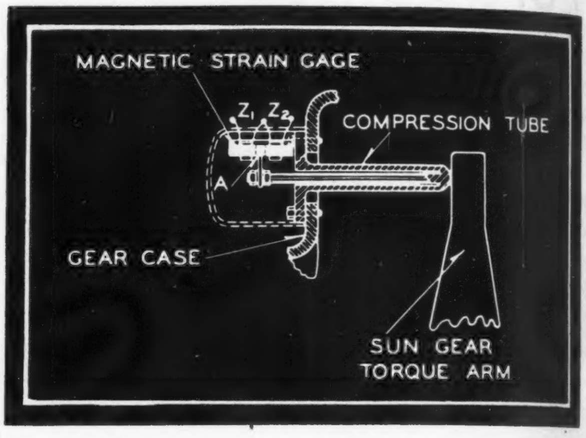
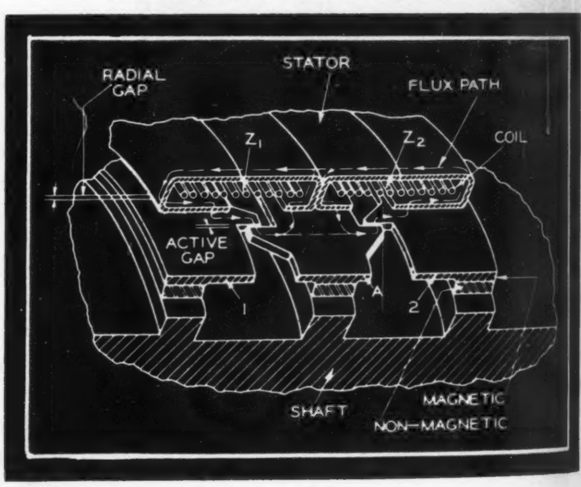


Fig. 4—Above—Magnetic strain gage shown applied to sun gear of planetary transmission. Compare with Fig. 1

Fig. 5—Below—Schematic view of magnetic coupled torquemeter, which does not require slip rings



in many precision gages or comparators as well as for aircraft engine testing (4).

It is usually small and operates on conventional alternating-current supply circuits, 60 cycles in the case of ground equipment and 400 cycles for airborne equipment, and can take any one of several different forms. The one shown in *Fig.* 3 with its connection diagram is simple and commonly used. It consists of two iron-core inductance coils with variable length air-gaps in their magnetic paths. Movement of the armature *A* of *Fig.* 3 to the left will decrease the air-gap of coil Z_1, increasing its inductance, and at the same time will increase the air-gap of coil Z_2 with a corresponding decrease of its inductance. Since Z_1 and Z_2 are both elements of a four-arm bridge, the output of the bridge indicated by the instrument *M* will be almost exactly proportional to the physical displacement of the armature *A* from its center position, within the use-

ful limits of the strain gage. The other two bridge arms, Z_3 and Z_4, are usually made part of a center-tapped autotransformer, which decreases the bridge impedance and thus increases the maximum signal energy available.

Magnetic strain gages can be mounted in the take-off head of the conventional hydraulic torquemeter, replacing the hydraulic piston and pressure gage. When this is done, a spring element, usually a short section of steel tubing, resists the reaction of the sun gear torque arm and the strain gage measures the deflections of the steel tube, *Fig.* 4. They also have been mounted directly upon the propeller shaft of an engine or reduction gear assembly with electrical connections made through slip rings on the shaft.

For shaft mounting, the slip-ring design is likely to be difficult from the standpoints of service trouble and contact resistance. On the other hand, there are numerous aircraft engine applications for torquemeters where it is desirable if not imperative to measure the twist in a shaft. The Magnetic Coupled torquemeter therefore was designed to eliminate slip rings and still retain a shaft-

Fig. 6—Magnetic coupled torquemeter for propeller shaft of an Allison engine employs many individual air gaps

mounted strain gage. A schematic illustration appears in *Fig.* 5 and a photograph of a typical assembly is reproduced in *Fig.* 6. Referring to ***Fig.* 5**, the shaft is provided with three flanges on which are mounted three toothed rings of magnetic material, but separated from contact with the steel shaft by means of nonmagnetic spacers made of a metal such as bronze or brass. Overlapping teeth or projections from each of the three magnetic rings form two sets of air gaps, one set between outer ring, 1, and the middle ring *A*, and one set between outer ring, 2, and the middle ring *A*. When the shaft is loaded or twisted in a clockwise direction, the air gaps between teeth on rings 1 and *A* are shortened and the air gaps between teeth on rings 2 and *A* are lengthened; this is in effect the same as moving the armature *A* of *Fig.* 3 to the left.

Magnetic flux flows across the two sets of air gaps, induced by the two stationary coils Z_1 and Z_2, also corresponding to Z_1 and Z_2 of *Fig.* 3. Coils Z_1 and Z_2 are shown completely encircling the shaft assembly and the magnetic return paths are through their encasing shells of magnetic material and across the radial air gaps between the stationary coil assembly and the rotating shaft assembly. These air gaps have relatively large effective areas and short lengths so that their magnetic permeances are large compared with the permeances of the active air gaps between the toothed rings; consequently the insertion of the radial air gaps in the magnetic circuit has only a small effect upon the characteristics of the strain gage. Normal eccentric movements of the rotating shaft in its bearings relative to the stator cause second-order changes in permeance of the radial air gaps and finally result in only third-order changes in torquemeter calibration.

When strain-gage elements are mounted on a rotating shaft, precautions must be taken to avoid errors due to bending moments and compression or tension loads on the shaft so that only pure torque is measured. This requires that a balanced magnetic circuit be used with both Z_1 and Z_2 responsive to shaft torque loads, and it also requires that the sensitive air gaps be distributed uniformly around the circumference of the shaft. Slip-ring installations of two conventional magnetic strain gages diametrically opposite each other on a shaft and with their coil circuits either connected in series or in parallel have shown a satisfactory lack of response to bending loads on the shaft where moderate loads are imposed. Some aircraft engine propeller shafts are subjected to bending moments from propeller gyroscopic couples in spins that may result in maximum fiber stresses of the order of 100,000 pounds per square inch where normal stress in torsion is of the order of 8,000 to 16,000 pounds per square inch.

Well distributed sets of sensitive air gaps are desirable in such circumstances and *Fig.* 6 shows a rotor for a Magnetic Coupled torquemeter with a large number of small individual air gaps in each set around the shaft. This particular shaft element is intended to be mounted between two existing flanges on an engine propeller shaft, and therefore in order to divide circumferential strain due to twist in the shaft length between flanges equally between the two sets of air gaps, three rotor rings are

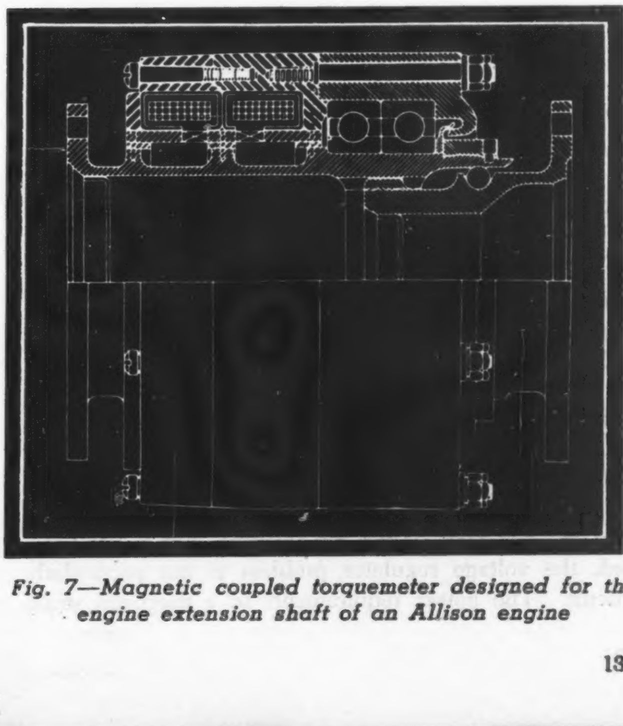


Fig. 7—Magnetic coupled torquemeter designed for the engine extension shaft of an Allison engine

mounted on a torsionally flexible cage the ends of which are permanently fastened to the two flanges. The stator carrying the two coils and their magnetic shells is also shown in the same illustration, and is bolted to the inner thrust bearing retainer cage inside of the propeller reduction-gear housing.

Another Magnetic Coupled torquemeter, intended for use on the extension shaft between the engine and a separate propeller reduction gearbox, is shown in *Fig.* 7. The three toothed magnetic rings are mounted on three shaft flanges instead of on a flexible cage, and the short shaft section carrying the rings is inserted in the extension shaft drive with flanged or splined couplings. A better mechanical job is obtained when it is possible to mount the toothed rings directly upon individual flanges integral with the shaft rather than upon the surface of a flexible cage or sleeve.

Effects of ambient temperature changes upon magnetic strain gages can be divided into those originating in the instrument circuit and those originating in the shaft pickup unit. Temperature effects in the instrument circuit

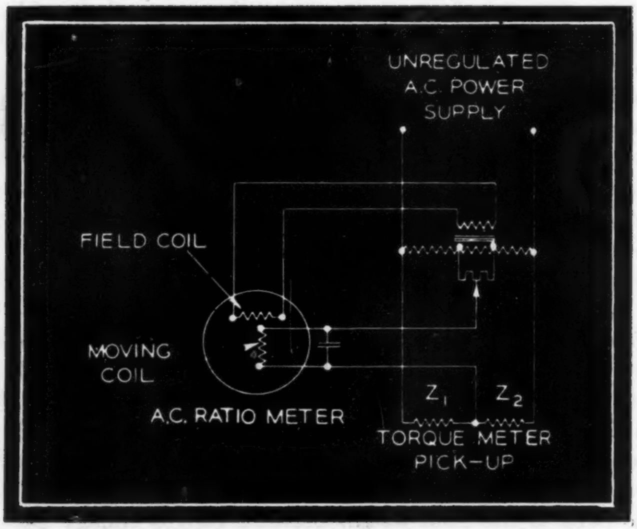


Fig. 8—Simplified electrical circuit for a magnetic strain gage, employing an alternating-current ratio meter

are readily corrected by conventional means and attention to good design practice. The temperature effects in the shaft element are more difficult to handle ordinarily, particularly since temperatures may range from -50 degrees Cent. to hot engine oil temperatures. By exercising care in design and application, these temperature effects can be held within acceptable limits.

A number of precautions must be observed in applying magnetic strain gages on Magnetic Coupled torquemeters to aircraft instrumentation problems. The alternating-current power supplies ordinarily available on airplanes may be subject to rather wide variations in voltage and frequency; and while close frequency regulation is not essential, the calibration of the torquemeter is proportional to the supply voltage when a simple circuit such as shown in *Fig.* 3 is used. Therefore a voltage-regulated power supply is essential to accurate calibration under these circumstances. Since the electrical load is small and fixed, the voltage regulator problem is not particularly difficult. The power requirements of a magnetic strain gage at 400 cycles is about five volt-amperes. The Magnetic Coupled torquemeter, due to greater iron loss and flux leakage requires thirty to fifty volt-amperes. Ordinary frequency variations are compensated for, when compensation is required, by the use of a small capacitor in the meter circuit. On airplanes where the only available electrical power is direct current, it is possible to use a small inverter to obtain the necessary 400-cycle power. Various regulators, such as the carbon-pile, saturated reactor, and centrifugal speed-governing type are available. By designing a unit specially for the job, it is possible to reduce the weight of the inverter, complete with regulator, to about two pounds per torquemeter on a two-engine airplane.

The indicating instrument usually is a rectifier-type milliammeter, and the complete bridge output circuit must include the proper external resistances to insure that the instrument circuit will provide accurate voltage measurements of the strain-gage bridge output over wide ranges of ambient temperature, input voltage, and frequency.

Use of a direct-current ratio meter instead of a conventional milliammeter eliminates the necessity for the voltage regulator, and since frequency variations are relatively unimportant in magnetic strain-gage instrument errors, its use has many attractive features. Unfortunately, the scale length of present direct-current ratio meters is limited to that of a 90-degree meter movement, and the scale is nonlinear.

Employs Simplified Circuit

A greatly simplified torquemeter utilizing the magnetic strain gage working directly into an alternating-current ratio meter is shown in the connection diagram of *Fig.* 8, in which voltage regulators, rectifiers, and all biasing circuits are eliminated. The stationary coil of the instrument is energized directly from the power supply source, and the moving coil is connected to the bridge output. Current is conducted to the moving coil through soft gold spirals with a minimum of spring restraint, and the moving element comes to rest when the in-phase component of the voltage induced in the coil is equal to and opposes the bridge output voltage. Since the power factor of the stationary coil is usually quite low, the out-of-phase voltages induced in the moving coil are small and have a negligible effect upon accuracy. Physically the alternating-current ratio meter movement resembles the conventional frequency meter movement. Close coupling between the moving coil and the stationary coil is desirable in order to reduce power requirements, and a laminated silicon steel or Hypernik magnetic structure is used. Long-scale or 270-degree alternating-current instrument movements have not yet been developed for aircraft use, and the alternating-current ratio meter airplane engine torquemeters will probably have to wait until such movements are available in production.

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Chaotic Conditions Would Follow Lag in Reconversion

RELEASE of workers as war contracts are terminated will have serious consequences unless steps are taken to speed up the War Production Board's four-point program for output of civilian goods. As an example of the number of men and women who are apt to be available for other work—or subject to unemployment—one has only to consider the Willow Run plan to cut production in half during the next six or eight months. It is estimated that, taking the aviation industry as a whole, as many as 294,000 employees will be released during this period—and this figure cannot fail to be very much higher if hostilities with Germany cease in the meantime.

Many of the released workers undoubtedly will be absorbed by other industries, such as shipbuilding, but here again there are bound to be cutbacks if the progress of the war continues as favorably as at present. Thus a vicious circle will be started, with no sound means to take up the slack other than civilian production.

An even more important consideration than the foregoing is the return of men from the Services in increasing numbers during the coming months, and the vast quantity of manpower that will become available after peace is declared—a quantity it would be impossible to absorb even under an ideal condition of total reconversion and high production.

Under those circumstances it is to be regretted that arrangements have been made for Donald M. Nelson, the sponsor of the four-point initial reconversion plan, to leave the country on an extended visit to China. It is to be hoped that during his absence nothing will occur to stymie the WPB "spot authorizations" for production of civilian goods—an idea for which Nelson should be credited.

It also is to be hoped that, while machine-building companies can feel no definite assurance at the moment that manpower and materials will be available to them in the near future, eligible concerns will lose no time in completing their plans and going after spot authorization to enable them to reconvert their plants—at least to some extent—as a prelude to full production later.

Civilians can get along without the goods but the country can't get along without high production!

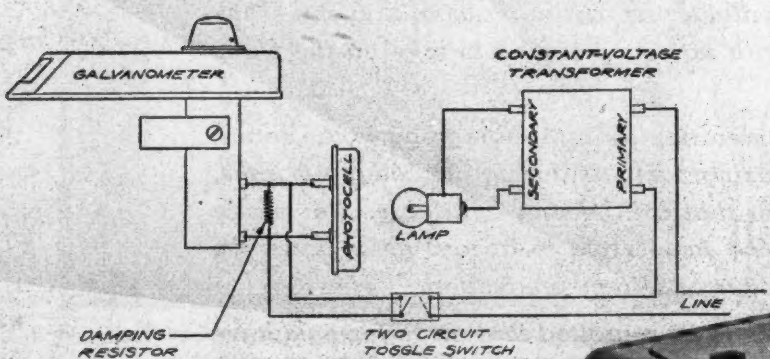
L. E. Jermy

Outstanding Designs

Compact Filter Photometer

Main elements of this electrical instrument made by Central Scientific Co. and known as the "Photelometer" include: A small lamp or light source, a shutter for controlling the amount of light passing through the system, a lens for concentrating the light beam, a glass cell for holding the solution under test, a photoelectric cell for measuring the amount of light passing through the sample, and a galvanometer for measuring the current produced by the photoelectric cell. Since the photoelectric cell measures only the light which strikes its sensitized surface and does not distinguish colors as such, a glass filter of proper color for transmitting only the color of light selected is required. For this purpose three colored filters are mounted in a sliding frame so that each one may be placed in the light beam as needed.

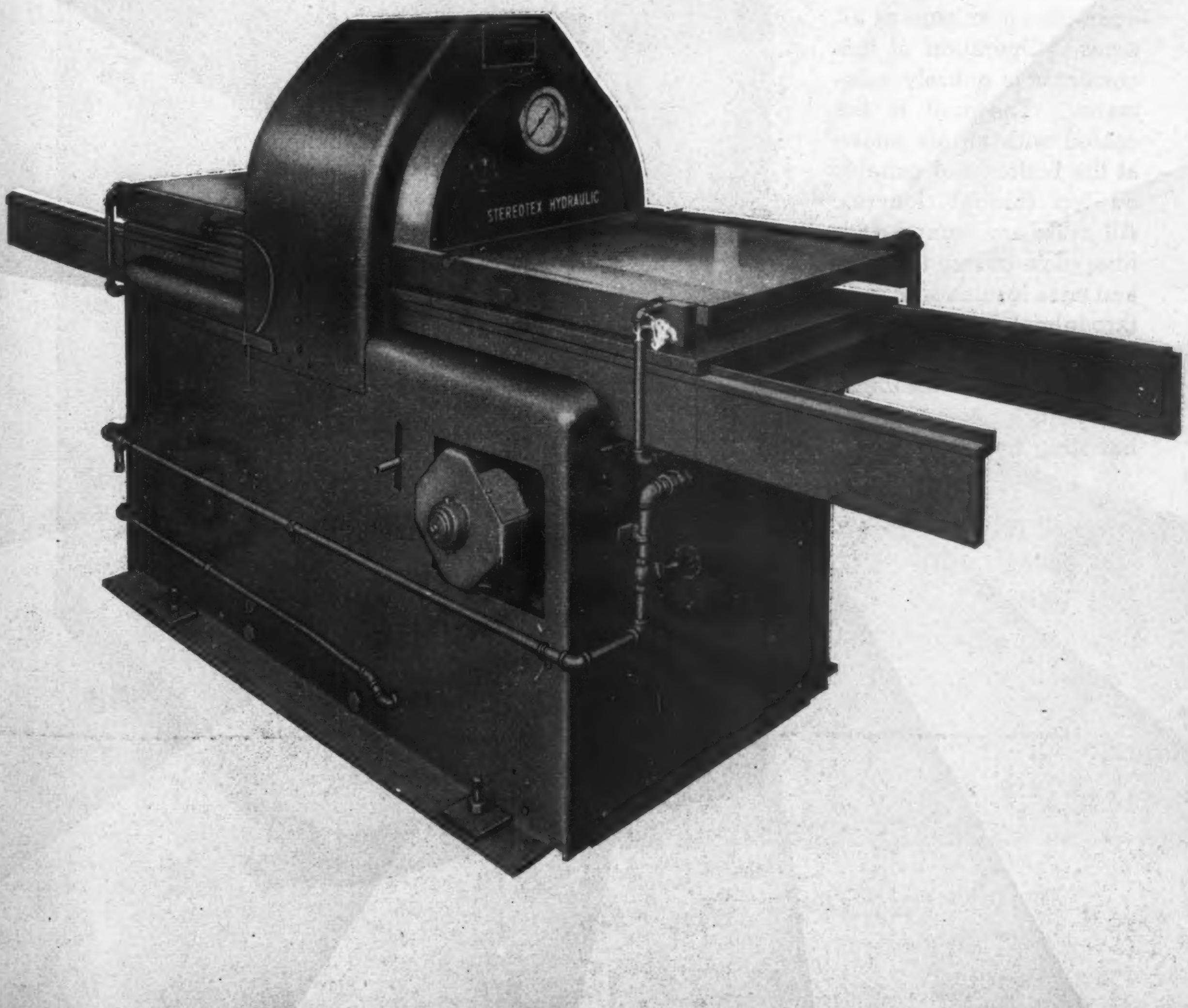
Voltage for operating the light source is controlled by a constant-voltage transformer which delivers a constant output to the lamp even when the power-supply line voltage varies as much as 15 per cent from normal. The controlling switch purposely is placed in the primary line of the transformer to permit a direct line to the lamp, thus avoiding the uncertainty of switch contacts in the lighting circuit and consequent varying light intensity.



Preheating Molding and Chilling Press

With fully automatic hydraulic system subject to pushbutton and automatic timer control, this press, manufactured by Stereotex Machinery Co., is used for making plastic electrotype molds, printing plates, matrices, etc. The hydraulic system consists essentially of the press proper, V-type rotary pump which is flange-mounted on a constant-speed motor, filling check valve, solenoid-actuated four-way valves, sequence valves, and a pressure-control valve. An air filter is provided on the completely enclosed oil reservoir and an oil filter on the suction line of the pump so that there is no possibility of dust or dirt getting into the oil or the hydraulic system. A cooler is provided inside the reservoir for controlling oil temperature.

Heating system employed is entirely self-contained and operated either by high-pressure gas or electricity. Both systems provide an accuracy of plus or minus 2 degrees Fahr. in the automatic temperature control of the lower and upper platens of the molding press and the preheating press, both of which are heated by superheated water generated by immersion-type heaters.



Transformer Arc-Welding Machine

This "all-weather" welder, manufactured by Wilson Welder and Metals Co. Inc., is of the high reactance type in which the output current is controlled by adjusting the reactance by moving the primary coil away from or nearer to the secondary coil. Center core of this shell-type construction is surrounded by the coils, the secondary coil being thoroughly anchored to the core, excluding all possibility of shifting. The primary coil is carried on a sturdy spider the center of which is threaded for an adjusting screw terminating in the control wheel.

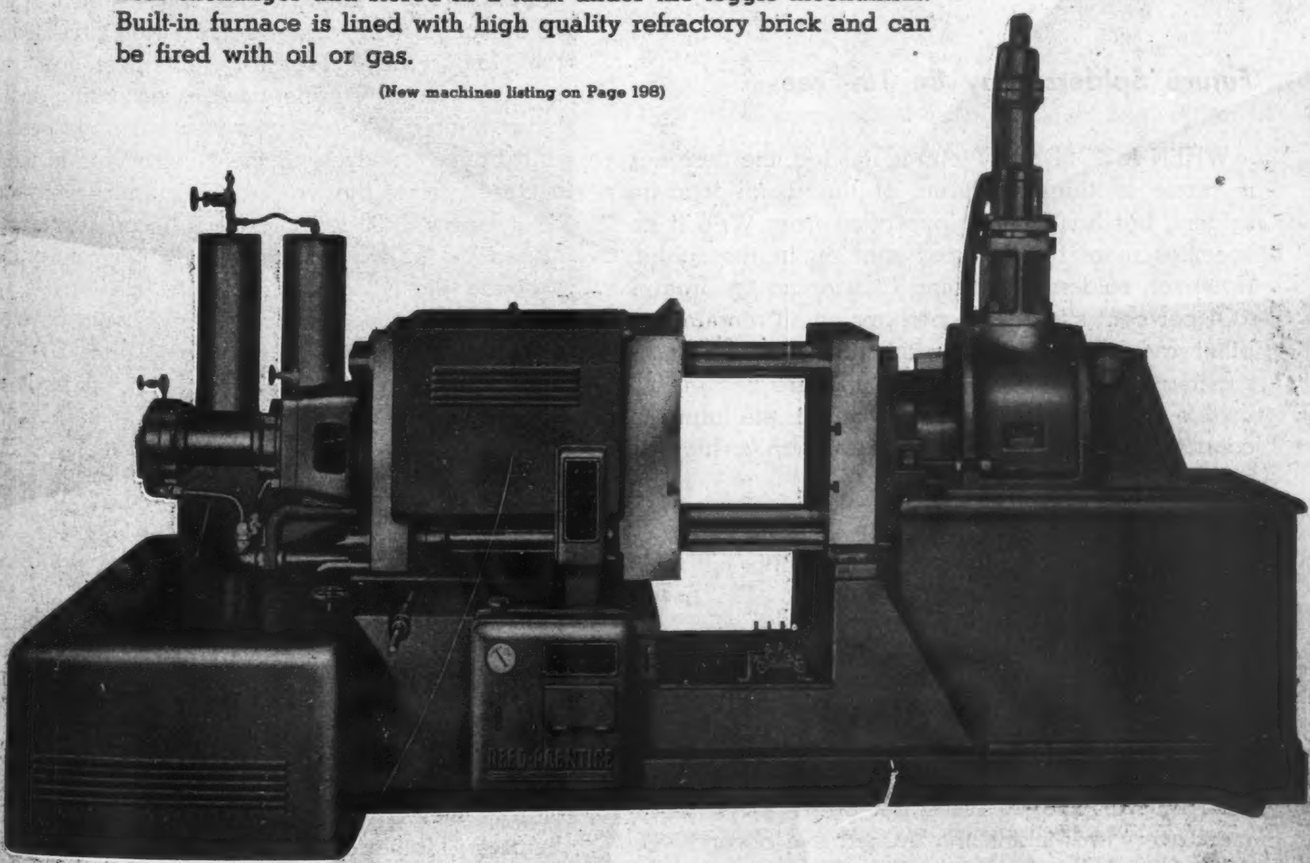
The secondary open-circuit voltage was selected as the lowest value for safety commensurate with good performance, and safety of the operator is further augmented by offering a low voltage contactor which holds the secondary voltage at less than half of the open-circuit voltage at all times. Operation of this contactor is entirely automatic. The unit is fan cooled with an air intake at the bottom and exhaust on top through louvres. All coils are wound with fiber-glass-covered wire, and mica insulation is used throughout. The case is easily removable for inspection and cleaning. Eye bolts on top facilitate handling by crane.

Hydraulic Die Casting Machine

Combination low and high-pressure vane pump employed in the hydraulic system of this die casting machine made by Reed-Prentice Corp. offers a maximum pressure of 1000 pounds per square inch and has capacity to recharge accumulator bottles in three seconds. Dies are closed by a hydraulically operated toggle mechanism composed of electric steel members connected with nitrided toggle pins moving in hardened steel bushings. Die plates are heavy steel castings with T-slots for the die-holding bolts and are supported by large steel tie bars which form the frame of the die locking mechanism.

Die plate and plunger movements are controlled manually by pushbuttons through solenoid-actuated four-way valves. From one position, the operator can start and stop the motor, operate the die plate and plunger, adjust the time clock to control the cooling cycle and adjust plunger pressure. The dies can be closed only by pushing the die-closing button with one hand and the safety button with the other. The plunger will shoot only when the safety interlock switch is closed and this occurs only when the dies are completely locked. The hydraulic oil utilized is cooled by a heavy-duty heat exchanger and stored in a tank under the toggle mechanism. Built-in furnace is lined with high quality refractory brick and can be fired with oil or gas.

(New machines listing on Page 198)



Design Roundup

Camera Makes Log

IN FLIGHT TESTS cameras are set up in the pilot's compartment to photograph the instrument panel. For this both still and movie cameras are used by Douglas. Formerly, a pilot took off with a board strapped to his knee to which a pad of paper was attached. He was supposed to record the antics of the maze of dial indicators before him and transfer this information to his report of the test flight. That in itself was a job to say nothing of the innumerable other chores he had to do to keep a green ship in the air. Now the camera does this book work for him, making an unquestionable record of the instrument indications and enabling the designers accurately to appraise the performance of the craft.

Future Solders May Be Tin-Free

WHEN A SOLDERED joint is needed, the designer is prone to think in terms of the usual lead-tin solders, but has to seek exception from WPB if he specifies more than 30 per cent tin in the solder. However, solders containing little or no tin, from 1 to 4 per cent silver, and perhaps small amounts of other available alloying elements, have strengths superior to the lead-tin solder and are proving so usable that the word solder is likely in the future to connote an almost tin-free rather than a high-tin alloy.

Design Ingenuity Will Set Pace

AMERICAN PRODUCTION OF WAR GOODS, phenomenal because of the tremendous volume attained in a period of less than three years, becomes all the more remarkable when one considers the amount of machinery, equipment and tools which had to be built before factories could get under way. In almost every war plant, the trained eye observes an abundance of machines and facilities that had to be designed specially to do specific jobs. In some instances, these operations were unknown in peacetime manufacture, or had been conducted on such small scale that methods used would not meet the demands of mass production. But American ingenuity was equal to the occasion, and design engineers developed the equipment required. Outstanding among these accomplishments are single-purpose machine tools, materials handling devices, testing and inspection apparatus, and the like. Under pressure of war, mechanical development has been swift, giving assurance that the approaching return to peace-time industry will be on a plane of efficiency, productivity and quality undreamed of before Pearl Harbor.

Trends in Diesel Design

DIESEL ENGINE DEVELOPMENT stimulated by the Navy since the last war has resulted in there being much more diesel horsepower than steam in the Navy today. Probably the most spectacular trend in diesel engine design, according to J. P. Stewart of B-W Superchargers Inc., has been the shift from the 4-cycle to the 2-cycle type in the last decade. Before the war 2-cycle engines were rarely built below 1000 horsepower. Recent statistics released by the Navy show that 75 per cent of all its installed diesel horsepower is of the 2-cycle type, much of it in small engines. Now being developed are 2-cycle engines ranging from 15 to 2500 horsepower for operation at higher piston speeds than the 4-cycle types.

Roundup Briefs

IN DESIGNING heavy equipment, shaved gears may be specified from a cost standpoint since gear makers have found it possible to apply the shaving process to sizes up to 18 feet in diameter with the same economies as in making smaller gears . . . More predictions are heard to the effect that single-purpose machine tools will find wider favor in post-war as well as standard tools with single-purpose fixtures. Motive: Longer, lower-cost runs of many products. Pressure to reduce costs is expected to be terrific!

Tables Facilitate Design of Rotating Disks

By William Knight

Curtiss-Wright Corp., Propeller Division

ROTATING disk wheels, of which turbine wheels are a typical application, are subject to the triple action of radial, tangential and axial stresses. Of the three, the radial and the tangential are the most important; the third can be neglected if no sudden change in thickness occurs at the hub.

In his mathematical analysis of disk wheels, Stodola assumed that the outline of the disk is a hyperbola and that the effect of the blades is equivalent to having their centrifugal forces uniformly distributed at the periphery of the disk. By making such assumptions it was possible to derive a formula for determining the radial and tangential stresses at any point between the bore and the periphery. However, the calculations must be repeated several times before the right proportions for securing the most efficient design can be determined. For this reason an attempt was made by the author to rewrite the Stodola formula and to put it in a shape that will allow its ready use and will yield the desired design data in a few minutes.

A disk of hyperbolic profile is such that the thickness y at any point distant x from the center is equal to:

$$y=\frac{c}{x^{\alpha}} \qquad (1)$$

where α can be positive, negative, or zero. If $\alpha=0$, $y=c$ for any value of α, which is the case of a disk of uniform thickness. If α is positive, the thickness of the disk is larger at the bore than at the periphery, which is the case with all turbine wheels.

If the section of a turbine wheel is tentatively laid out, the values of c and α that will give values of y for various values of x, closely approaching the thickness of the wheel may be calculated. Thus

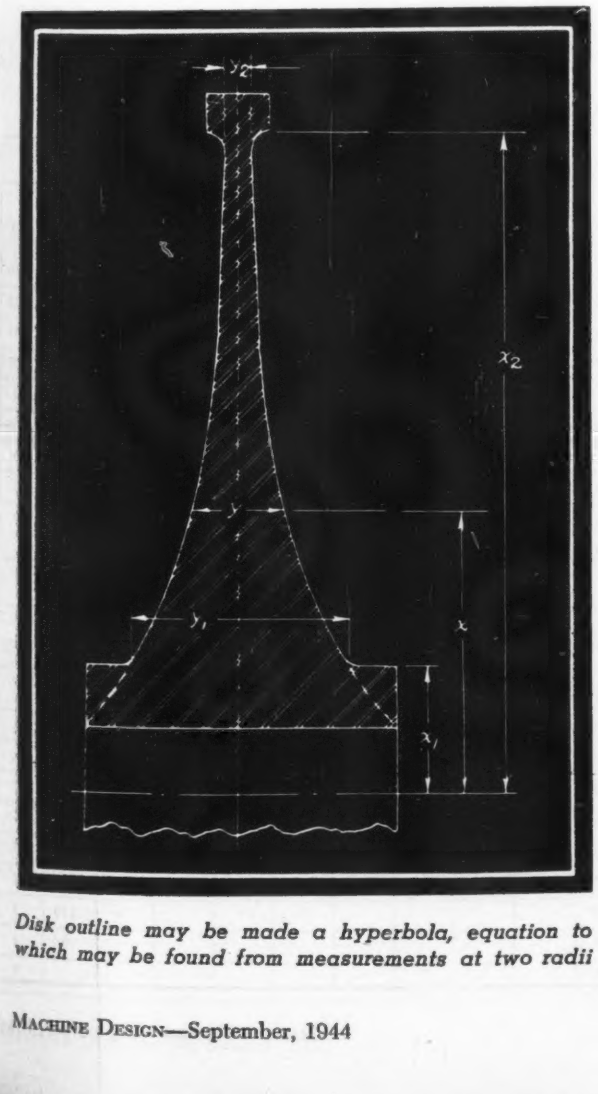
$$\alpha=\frac{\log\ y_1-\log\ y_2}{\log\ x_2-\log\ x_1} \qquad (2)$$

in which y_1, and y_2 are the thicknesses of the wheel at two points distant x_1 and x_2 from the center, as in the accompanying figure. Having found α, $c=y_1x_1{}^{\alpha}=y_2x_2{}^{\alpha}$. All the points of a hyperbola that will coincide with the outline of the wheel at x_1 and x_2 can then be found. The outline of the wheel can be slightly modified so as to make it follow the hyperbolic profile as far as possible, or a hyperbola that will fit more closely to the outline of the wheel as laid out may be found.

It is well to note that, although a variation of the hyperbola from the outline of the disk as laid out, in points near the bore, will have only a small influence on the calculated stresses, any variation in points near the periphery will have a decided influence.

Having found the value of α giving a hyperbola that will closely approach the outline of the wheel, the

This data sheet is based on a paper presented at a recent meeting of the Society of Automotive Engineers.



Disk outline may be made a hyperbola, equation to which may be found from measurements at two radii

ENGINEERING DATA SHEET

values of σ_r and σ_t (radial and tangential stresses due to the centrifugal force of the rotating disk only) and σ_r' and σ_t' (radial and tangential stresses due to the centrifugal force of the projecting masses at the periphery of the disk) can be obtained from TABLES I to V. The stresses are given as functions of σ (the tangential stress of a thin ring rotating at the same peripheral speed as the disk) and σ_3 (the centrifugal force of projecting masses at the periphery, pounds per square inch), thus:

$\sigma_r = \sigma n_r =$ Radial stress due to disk only

$\sigma_t = \sigma n_t =$ Tangential stress due to disk only

$\sigma_r' = \sigma n_r' =$ Radial stress due to projecting masses

$\sigma_t' = \sigma n_t' =$ Tangential stress due to projecting masses.

Values of the stress factors are given for selected values of m and m_0 where:

$$m = \frac{\text{Diameter at any point}}{\text{Outside Diameter of disk}}$$

$$m_o = \frac{\text{Diameter at bore}}{\text{Outside Diameter of disk}}$$

The general formulas for σ_r σ_t σ_r' and σ_t' are:

$$\sigma_r = \sigma \left\{ A \left[-m^2 + \frac{m^{\Psi_1-\Psi_2}(m_0^{3-\Psi_2}-1) - m_0^{3-\Psi_2} + m_0^{\Psi_1-\Psi_2}}{m^{1-\Psi_2}(m_0^{\Psi_1-\Psi_2}-1)} \right] \right\} \quad (3)$$

$$\sigma_t = \sigma \left\{ B \left[-m^2 + \frac{Cm^{\Psi_1-\Psi_2}(m_0^{3-\Psi_2}-1) + D(m_0^{3-\Psi_2} - m_0^{\Psi_1-\Psi_2})}{m^{1-\Psi_2}(m_0^{\Psi_1-\Psi_2}-1)} \right] \right\} \quad \ldots (4)$$

$$\sigma_r' = \sigma_3 \frac{m_0^{\Psi_1-\Psi_2} - m^{\Psi_1-\Psi_2}}{m^{1-\Psi_2}(m_0^{\Psi_1-\Psi_2}-1)} \quad \ldots\ldots (5)$$

$$\sigma_t' = \sigma_3 \frac{\psi_1 m_0^{\Psi_1-\Psi_2} - \psi_2 m^{\Psi_1-\Psi_2}}{m^{1-\Psi_2}(1-m_0^{\Psi_1-\Psi_2})} \quad \ldots\ldots (6)$$

where

$$A = (1+3\nu)(1-\nu^2)B = 1.73B$$

$$B = \frac{1+3\nu}{8-(3+\nu)\alpha} = \frac{1.9}{8-3.3\alpha}$$

$$C = \frac{A}{B}\psi_2 = 1.73\psi_2$$

$$D = \frac{A}{B}\psi_1 = 1.73\psi_1$$

$\nu =$ Poisson's ratio (assumed $= .3$)

$$\psi_1 = \frac{\alpha}{2} + \sqrt{\frac{\alpha^2}{4} + \alpha\nu + 1}$$

$$\psi_2 = \alpha - \psi_1$$

Table I—Stress Factors for Shape Constant $\alpha = 2$

Tangential, n_t, due to disk

m_o	m = .1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	.07	.08	.097	.118	.135	.145	.149	.131	.122	.082
.2		.262	.192	.180	.181	.180	.177	.157	.148	.105
.3			.414	.321	.278	.256	.239	.213	.191	.152
.4				.550	.443	.376	.336	.292	.257	.203
.5					.668	.550	.465	.401	.353	.291
.6						.760	.643	.553	.478	.399
.7							.850	.735	.646	.545
.8								.903	.785	.675
.9									.95	.84
1.0										1 00

Tangential, n_t', due to projecting masses

m_o	m = .1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	.077	.066	.099	.147	.206	.272	.349	.429	.522	.616
.2		.240	.188	.204	.246	.303	.373	.451	.539	.633
.3			.466	.380	.371	.398	.451	.514	.596	.684
.4				.768	.645	.607	.617	.654	.717	.791
.5					1.172	1.012	.940	.925	.952	1.00
.6						1.71	1.526	1.395	1.355	1.365
.7							2.716	2.425	2.170	2.09
.8								4.344	3.91	3.65
.9									9.25	8.42
1.0										∞

Radial, n_r, due to disk

m_o	m = .1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	0	.069	.118	.153	.174	.178	.164	.124	.069	0
.2		0	.084	.130	.160	.166	.156	.120	.069	0
.3			0	.083	.127	.149	.144	.110	.069	0
.4				0	.074	.113	.128	.100	.065	0
.5					0	.059	.088	.080	.052	0
.6						0	.038	.051	.031	0
.7							0	.017	.013	0
.8								0	.004	0
.9									0	0
1.0										0

Radial, n_r', due to projecting masses

m_o	m = .1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	0	.067	.139	.225	.325	.437	.561	.696	.842	1
.2		0	.105	.204	.312	.427	.555	.692	.837	1
.3			0	.141	.270	.398	.538	.680	.835	1
.4				0	.178	.338	.498	.657	.826	1
.5					0	.217	.417	.608	.803	1
.6						0	.270	.519	.757	1
.7							0	.357	.688	1
.8								0	.510	1
.9									0	1
1.0										1

Table II—Stress Factors for Shape Constant $\alpha=1.5$

Tangential, n_t, due to disk

m_o \ m	.1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	.174	.125	.140	.158	.172	.178	.173	.158	.133	.101
.2		.351	.255	.229	.224	.216	.204	.182	.157	.121
.3			.490	.356	.312	.284	.259	.228	.196	.167
.4				.610	.480	.417	.363	.311	.269	.223
.5					.708	.584	.498	.421	.367	.308
.6						.792	.664	.558	.489	.419
.7							.86	.72	.634	.545
.8								.905	.775	.673
.9									.95	.830
1.0										1.0

Tangential, n_t', due to projecting masses

m_o \ m	.1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	.192	.148	.186	.243	.310	.373	.446	.518	.596	.673
.2		.440	.332	.341	.374	.423	.485	.552	.626	.698
.3			.713	.571	.542	.556	.587	.633	.698	.764
.4				1.006	.873	.750	.710	.751	.855	.898
.5					1.469	1.251	1.152	1.100	1.110	1.131
.6						1.930	1.775	1.613	1.565	1.535
.7							2.93	2.556	2.4	2.28
.8								4.6	4.1	3.85
.9									9.67	8.71
1.0										∞

Radial, n_r, due to disk

m_o \ m	.1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	0	.101	.154	.189	.194	.198	.178	.140	.087	0
.2		0	.104	.161	.180	.187	.173	.133	.079	0
.3			0	.095	.142	.156	.149	.118	.067	0
.4				0	.057	.113	.123	.101	.06	0
.5					0	.061	.088	.082	.053	0
.6						0	.051	.06	.04	0
.7							0	.035	.031	0
.8								0	.022	0
.9									0	0
1.0										0

Radial, n_r', due to projecting masses

m_o \ m	.1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	0	.131	.234	.335	.44	.557	.656	.767	.881	1
.2		0	.170	.298	.417	.532	.646	.764	.883	1
.3			0	.197	.353	.49	.619	.747	.87	1
.4				0	.228	.407	.565	.717	.86	1
.5					0	.257	.47	.663	.829	1
.6						0	.31	.57	.787	1
.7							0	.403	.713	1
.8								0	.535	1
.9									0	1
1.0										1

Table III—Stress Factors for Shape Constant $\alpha=1$

Tangential, n_t, due to disk

m_o \ m	.1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	.295	.201	.205	.216	.229	.229	.207	.189	.159	.12
.2		.463	.349	.305	.286	.271	.244	.229	.187	.155
.3			.584	.438	.379	.346	.303	.268	.231	.185
.4				.680	.536	.463	.399	.350	.304	.25
.5					.759	.620	.530	.455	.395	.330
.6						.819	.690	.586	.512	.445
.7							.865	.739	.637	.55
.8								.909	.788	.69
.9									.950	.835
1.0										1.0

Tangential, n_t', due to projecting masses

m_o \ m	.1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	.456	.322	.353	.408	.470	.539	.600	.654	.703	.781
.2		.764	.571	.540	.559	.604	.650	.694	.735	.797
.3			1.058	.835	.759	.749	.760	.782	.807	.882
.4				1.315	1.180	.982	.940	.924	.923	1.023
.5					1.750	1.500	1.400	1.304	1.252	1.290
.6						2.25	2.00	1.846	1.711	1.715
.7							3.186	2.846	2.576	2.49
.8								4.926	4.346	4.10
.9									9.76	9.00
1.0										∞

Radial, n_r, due to disk

m_o \ m	.1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	0	.146	.204	.227	.235	.231	.187	.141	.078	0
.2		0	.126	.183	.209	.214	.176	.133	.076	0
.3			0	.116	.166	.188	.161	.122	.072	0
.4				0	.097	.145	.134	.109	.068	0
.5					0	.083	.095	.088	.058	0
.6						0	.051	.061	.046	0
.7							0	.031	.03	0
.8								0	.016	0
.9									0	0
1.0										0

Radial, n_r', due to projecting masses

m_o \ m	.1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	0	.249	.383	.490	.589	.692	.763	.845	.926	1
.2		0	.264	.421	.548	.665	.746	.833	.921	1
.3			0	.270	.453	.605	.706	.812	.912	1
.4				0	.283	.499	.636	.775	.893	1
.5					0	.325	.525	.711	.87	1
.6						0	.335	.597	.813	1
.7							0	.410	.735	1
.8								0	.560	1
.9									0	1
1.0										1

Engineering Data Sheet

Table IV—Stress Factors for Shape Constant $\alpha = .5$

Tangential, n_t, due to disk

m_o	m = .1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	.5	.323	.304	.299	.29	.284	.255	.226	.191	.145
.2		.617	.445	.392	.349	.32	.291	.255	.217	.175
.3			.706	.55	.465	.418	.364	.317	.27	.22
.4				.763	.619	.527	.45	.397	.335	.27
.5					.818	.662	.575	.493	.425	.355
.6						.853	.733	.63	.55	.475
.7							.905	.765	.659	.57
.8								.934	.811	.7
.9									.968	.84
1.0										1.0

Tangential, n_t', due to projecting masses

m_o	m = .1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	.985	.644	.63	.657	.691	.722	.757	.788	.820	.849
.2		1.257	.933	.844	.820	.819	.838	.853	.878	.900
.3			1.54	1.205	1.071	1.012	.988	.984	.988	1.02
.4				1.735	1.5	1.335	1.248	1.195	1.172	1.162
.5					2.19	1.86	1.67	1.55	1.485	1.445
.6						2.58	2.41	2.17	2.025	1.925
.7							3.55	3.12	2.85	2.66
.8								5.19	4.63	4.28
.9									10.14	9.19
1.0										∞

Radial, n_r, due to disk

m_o	m = .1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	0	.212	.264	.278	.267	.252	.202	.145	.079	0
.2		0	.169	.225	.235	.232	.188	.137	.076	0
.3			0	.135	.183	.202	.174	.132	.075	0
.4				0	.099	.150	.139	.107	.063	0
.5					0	.088	.101	.088	.053	0
.6						0	.038	.06	.042	0
.7							0	.034	.031	0
.8								0	.021	0
.9									0	0
1.0										0

Radial, n_r', due to projecting masses

m_o	m = .1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	0	.45	.601	.698	.767	.844	.878	.92	.962	1
.2		0	.406	.594	.709	.814	.863	.915	.969	1
.3			0	.375	.58	.735	.813	.89	.948	1
.4				0	.357	.6	.728	.837	.933	1
.5					0	.385	.605	.77	.903	1
.6						0	.375	.645	.855	1
.7							0	.44	.76	1
.8								0	.58	1
.9									0	1
1.0										1

Table V—Stress Factors for Disk of Uniform Thickness, $\alpha = 0$

Tangential, n_t, due to disk

m_o	m = .1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	.815	.493	.434	.408	.369	.337	.304	.266	.225	.181
.2		.823	.584	.488	.432	.385	.343	.299	.254	.206
.3			.830	.636	.533	.461	.404	.347	.299	.246
.4				.841	.675	.568	.49	.434	.362	.303
.5					.858	.706	.602	.517	.444	.375
.6						.876	.739	.634	.545	.467
.7							.897	.766	.66	.57
.8								.927	.803	.697
.9									.959	.837
1.0										1.00

Tangential, n_t', due to projecting masses

m_o	m = .1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	2.02	1.263	1.122	1.073	1.051	1.038	1.031	1.026	1.023	1.02
.2		2.08	1.505	1.3	1.207	1.156	1.125	1.105	1.091	1.082
.3			2.2	1.72	1.496	1.375	1.302	1.255	1.222	1.199
.4				2.38	1.955	1.72	1.58	1.489	1.426	1.381
.5					2.68	2.27	2.02	1.86	1.75	1.67
.6						3.12	2.8	2.44	2.25	2.12
.7							3.92	3.46	3.14	2.92
.8								5.56	4.98	4.56
.9									10.56	9.54
1.0										∞

Radial, n_r, due to disk

m_o	m = .1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	0	.294	.33	.322	.294	.254	.204	.145	.0775	0
.2		0	.206	.257	.257	.232	.192	.138	.0735	0
.3			0	.151	.198	.196	.17	.126	.0694	0
.4				0	.111	.144	.141	.11	.0625	0
.5					0	.079	.103	.091	.0543	0
.6						0	.0572	.0662	.0441	0
.7							0	.0346	.0305	0
.8								0	.0175	0
.9									0	0
1.0										0

Radial, n_r', due to projecting masses

m_o	m = .1	.2	.3	.4	.5	.6	.7	.8	.9	1.0
.1	0	.66	.9	.95	.97	.985	.99	.995	.999	1
.2		0	.58	.78	.88	.93	.96	.98	.99	1
.3			0	.48	.71	.825	.895	.945	.98	1
.4				0	.38	.66	.8	.89	.96	1
.5					0	.41	.655	.815	.925	1
.6						0	.415	.685	.87	1
.7							0	.46	.775	1
.8								0	.585	1
.9									0	1
1.0										1

ACHINE DESIGN'S

Materials Work Sheet

Filing Number 19.00

Zinc-Base Die Casting Alloys

ASTM No. B86-43
Alloys Nos. XXI, XXIII, XXV

AVAILABLE IN: Ingots for die casting

ANALYSES:

	Alloy XXI	Alloy XXIII	Alloy XXV
Aluminum	3.5-4.5	3.5-4.3	3.5-4.3
Copper	2.5-3.5	.1 max	.75-1.25
Magnesium	.02-.1	.03-.08	.03-.08
Iron, max	.1	.1	.1
Lead, max	.007	.007	.007
Cadmium, max	.005	.005	.005
Tin, max	.005	.005	.005
Zinc	remainder	remainder	remainder

PROPERTIES*

TENSILE STRENGTH
(psi)

	Alloys		
	XXI	XXIII	XXV
As cast	52,100	41,000	47,600
After 10 days in 95 C steam	44,100	34,300	37,600
After 2 years dry aging at 95 C	39,900	31,000	35,200
After 7 years of indoor aging	48,100	33,800	38,300

EXPANSION
(in. per 6 in.)

	Alloys		
	XXI	XXIII	XXV
As cast	.....	.....	.....
After 10 days in 95 C steam	.0084	.003	.0028
After 2 years dry aging at 95 C	.0195	.0013	.0065
After 7 years of indoor aging	.0055	.0006	.0004

TENSILE ELONGATION
(per cent in 2 inches)

	Alloys		
	XXI	XXIII	XXV
As cast	8	10	7
After 10 days in 95 C steam	3	14	4
After 2 years dry aging at 95 C	3	17	13
After 7 years of indoor aging	5	16	15

CHARPY IMPACT STRENGTH
(¼ by ¼-in. bars, unnotched, ft-lbs)

	Alloys		
	XXI	XXIII	XXV
As cast	35	43	48
After 10 days in 95 C steam	2	35	9
After 2 years dry aging at 95 C	1	36	7
After 7 years of indoor aging	7	41	42

PHYSICAL CONSTANTS

	Alloys		
	XXI	XXIII	XXV
Brinell Hardness (500-kg load, 10-mm ball)	100	82	91
Compressive Strength (psi)	93,000	60,000	87,000
Electrical Conductivity (per cent of Int'l. Annl'd Copper Std.)	25.23	27.16	26.5
Melting Point (deg C)	379.5	380.9	380.6
Melting Point (deg F)	715.1	717.6	717.1
Modulus of Rupture (psi)	116,000	95,000	105,000
Shearing Strength (psi)	46,000	31,000	38,000
Solidification Point (deg C)	379.3	380.6	380.4
Solidification Point (deg F)	714.7	717.1	716.7
Solidification Shrinkage (in./ft)	.15	.14	.14
Specific Gravity	6.7	6.6	6.7
Specific Heat (cal/g/deg C)	.1	.1	.1
Thermal Conductivity (cal/sec/sq cm/cm/deg C) at 18 C	.25	.27	.26
Thermal Expansion (per deg C)	.0000277	.0000274	.0000274
Thermal Expansion (per deg F)	.0000154	.0000152	.0000152
Transverse Deflection (in.)	.22	.27	.16
Weight (lb per in.³)	.24	.24	.24

MACHINE DESIGN is pleased to acknowledge the collaboration of The New Jersey Zinc Co. in this presentation.

*From tests conducted by The New Jersey Zinc Co. on specimens cast in dies of most improved design using latest casting technique.

Materials Work Sheet

CHARACTERISTICS

While these three alloys are similar in general properties, each has been formulated to emphasize some specific property or combination of properties. Alloy XXI emphasizes hardness and tensile strength. Alloy XXIII, compared to the other two, retains impact strength better and has better dimensional stability. Alloy XXV combines most of the strength of Alloy XXI with dimensional stability and impact strength approaching that of Alloy XXIII. Both Alloy XXI and XXV have excellent resistance to corrosion under severe atmospheric exposure.

APPLICATIONS

In a broad sense it may be said that die castings are suited best for machine parts which would be excessively expensive if made as built-up stamped and formed units or as mass-produced sand castings requiring a good deal of machining. Parts which have been die cast successfully of these alloys are veritably legion and a listing approaching completeness would be of inordinate length. For the sake of brevity the following typical applications are offered as indicative of the entire field:

Housings for sanders, band saws, air compressors, spray guns, electric drills, gears, grinders, etc. Housings, frames, gears, cams, levers, pulleys, links, etc., for all types of business machines such as typewriters, cash registers, calculators, etc. Also similar parts on automobiles, textile machines, coal stokers, pumps, meters, telephones, vacuum cleaners, washing machines, ironers, oil burners, cameras, etc.

FABRICATION

It is generally recognized that one of the primary advantages of using die cast parts lies in their need for comparatively little or no machining. For many applications, holes and bearing and mating surfaces "as cast" prove adequate without additional processing. However, in cases where the degree of accuracy in size or the quality of surface finish in the "as-cast" state is not sufficient, various kinds of machining are utilized.

MACHINABILITY:

All three of these alloys have good machinability and are comparatively soft. In general, high speeds and light cuts yield best results. Ordinary hardened high-carbon steel tools are satisfactory for most work although high-speed steel, Stellite and carbide tools often are used to advantage. While most machining is done dry, for particular work and most deep drilling and tapping, a lubricant makes for faster, smoother cutting, better finish and tends to promote accuracy and reduce tool breakage.

DRILLING:

High speeds and slow feeds are recommended. Speeds generally range from 200 to 300 feet per minute surface speed for high-speed steel drills and 100 to 150 feet per minute for carbon steel drills. Common practice is to use carbon steel drills for shallow holes and high-speed steel drills for deep holes. Standard drills ground to the standard point included angle of 118 degrees and having a lip clearance of at least 12 degrees are satisfactory. To prevent drills from "grabbing", often small flats are ground along the cutting edges, producing a zero rake angle. Use of lubricant during drilling is desirable but not absolutely essential.

REAMING:

Keen-edged, deep-fluted standard reamers having six straight flutes and narrow lands are used commonly for machine reaming although inserted-blade type reamers are satisfactory if rigid. Wide lands tend to produce burnishing and generate heat. In general, land width should not exceed .015-inch. Flute faces usually are ground to zero rake and the clearance angle back of the land generally approximates 10 degrees. If spiral-fluted reamers are employed, the spiral should be pronounced. Rose reamers are not recommended.

TAPPING:

Where standard threads of fine or medium pitch are required it is, in most cases, less expensive to tap than to cast the thread around a core which subsequently must be unscrewed. However, many times when coarse pitch threads or threads of special shapes are needed, it is more economical to cast them. When specifying the size of cored holes for tapping, it is well to bear in mind that core pins require some taper. Thus, the diameter at the small end of the hole must be large enough not to "squeeze" the tap, causing breakage, while at the opposite end the hole must not be excessively large. Generally the thread at the large end should be no less than 75 per cent of full depth. Where difference in hole diameter due to core-pin taper is excessive, as with deep holes, it is best to specify drilling or reaming to remove the taper in the hole before tapping.

In the majority of cases standard taps are satisfactory. Ample chip clearance must be provided. For holes up to about ⅜-inch diameter, two-fluted taps are preferred, while for

(*Continued on Page* 152)

PROPERTIES AT SUBZERO AND ELEVATED TEMPERATURES*

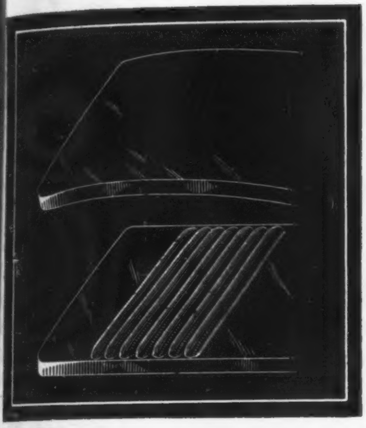
	Testing Temperatures					
	−40 C −40 F	−20 C −4 F	0 C 32 F	21 C 70 F	40 C 104 F	95 C 203 F
Tensile Strength (psi)						
Alloy XXIII	44,800	43,700	41,300	41,000	35,500	28,300
Alloy XXV	48,900	49,400	48,300	47,600	42,900	35,100
Elongation in 2 inches (per cent)						
Alloy XXIII	3	4	6	10	16	30
Alloy XXV	2	3	6	7	13	23
Charpy Impact Strength (¼ by ¼ in. bars, unnotched, ft-lbs)						
Alloy XXIII	2	4	23	43	42	40
Alloy XXV	2	4	41	48	46	43
Brinell Hardness (500-kg load, 10-mm ball)						
Alloy XXIII	91	87	82	82	68	48
Alloy XXV	107	104	99	91	89	62

*From tests conducted by The New Jersey Zinc Co. on specimens cast in dies of most improved design using latest technique.

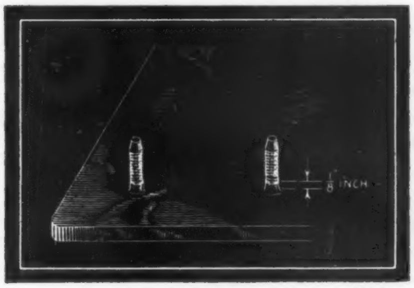
Materials Work Sheet

DESIGN TIPS

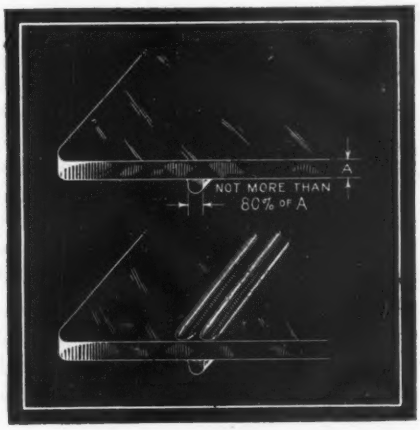
Plain Flat Areas—Left—When of considerable size, plain flat areas required to be cast with an exceptionally smooth surface such as would be suitable for plating are difficult to produce, lead to high rejections and attendant increased costs. Elimination of this casting problem without introducing finishing difficulties may be effected by slightly curving the flat areas or by breaking the surface with some simple design such as in shown in the illustration.

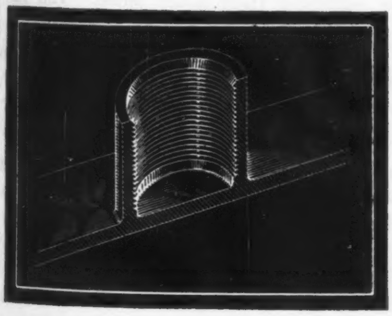


Studs—Right Center—Being less costly, studs as an integral part of a casting are preferable to inserted studs. However, to avoid damage in handling they should be cast to as large a diameter and as short a length as well-balanced design will permit. Unthreaded studs should be used in conjunction with speed fasteners wherever possible because they are more economical to cast and possess higher impact strength than threaded studs. If threaded studs must be employed they can be strengthened considerably by using ⅛-inch fillets at their bases. Also, it is good practice to allow about ⅛-inch from the end of the thread to the stud base and, to facilitate handling and assembly, the stud tips should be chamfered.



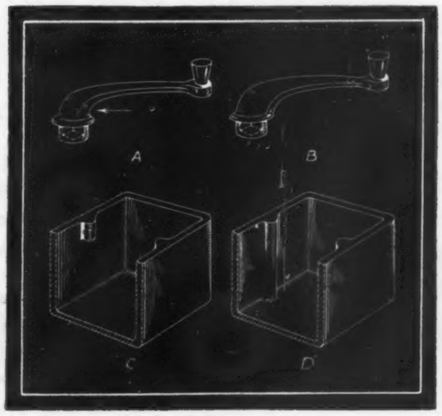
Shadow Marks — Left — Often where smooth surfaces of thin sections are backed up by ribs, bosses or studs, blemishes known as "shadow marks" appear on the surface directly opposite the protrusions. These shadow marks can be prevented by keeping the thickness of the casting above .1-inch or by designing the width of the underlying protrusion to a dimension no greater than 80 per cent of the wall thickness. Where these methods are not feasible a simple external design directly opposite the protrusion may be used as an effective camouflage. One such design is shown.

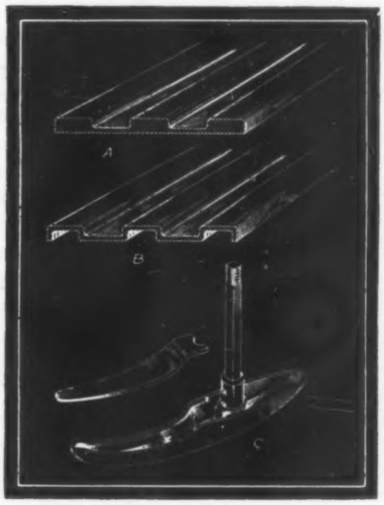




Bosses — Above — Because external threads, through notch effect, lower the impact strength of studs, tapped bosses are stronger than and almost always more desirable than threaded studs. About ⅛-inch should be allowed at the bottom of the tapped hole for chip clearance. If this is not done, frequent tap breakage will result. Countersink should be about 1/32-inch larger than the outside diameter of the thread to facilitate tapping and assembly.

Undercuts—Below—Generally are so costly that every effort should be made to avoid them. Views *A* and *B* show how undercut was eliminated by redesigning hub section of hand lever. Views *C* and *D* show how, by extending screw boss to base wall of housing, undercutting was made unnecessary.





Metal-Saving Cores — Above —Many times significant cost savings can be realized by eliminating excessive metal in die castings through judicious coring. However, such coring is justified only when the saving in metal does not weaken the casting or adversely affect the ease with which the part can be finished. Views *A* and *B* illustrate a part in which such coring was justified. In the part shown in view *C*, considerable metal was saved by coring and springing a die-cast cap over the bottom of a hollow handle.

Materials Work Sheet

TAPPING: *(Con'd.)*

the larger sizes, taps having one flute less than standard often are preferred. When tapping "through" holes not more than two diameters deep, spiral-pointed taps give good results. Regular straight-fluted taps are recommended for blind holes. As is the case with drilling, lubrication is desirable but not absolutely essential.

It should be added that tapping sometimes may be dispensed with entirely by using self-tapping screws which work quite satisfactorily with these die casting alloys.

SPOT FACING:

Tools having an odd number of large teeth with plenty of chip clearance between them are recommended. For long production runs carbide-tipped tools are desirable. Teeth should be ground with no rake and a clearance angle of about 12 degrees.

PUNCHING:

When properly supported by lower die, casting walls up to ⅛-inch thick may be punched and sheared often at higher production rates than are possible with drilling. It is common practice to remove flash and casting fins by punching.

BENDING AND FORMING:

These alloys are sufficiently ductile to permit bending and forming. Thus where it is not practical to cast a part to the desired shape, final forming often may be effected in a suitable die. Bending and forming operations on castings made of Alloy XXI should be performed immediately after casting because it is then in its most ductile state. Alloys XXIII and XXV may be bent or formed with equal ease any time after casting. If it is desired to make the metal more ductile, mild heating from 150 to 212 degrees Fahr. may help but seldom is necessary. In any case bending and forming should not be attempted at temperatures below 70 degrees Fahr.

FINISHES

PLATING:

The following is offered merely to acquaint the reader with the types of coatings which are being applied successfully and, in general, basic data concerning commercial procedure. It is not intended to serve as a basis for specification.

All of these alloys will take platings of brass, copper, nickel, chromium, gold and silver. Quality of finish depends largely on preparation of the part prior to applying the plating. Thus, to insure the high degree of luster desired, it is necessary, after trimming the castings of parting fins and gate stubs, to polish or buff the surfaces requiring plating. Proper cleaning after polishing or buffing is an absolute "must", for without it adherence of the plating will be impaired. Preliminary removal of grease often is done in a degreasing unit using trichlorethylene. This generally is followed by alkaline cleaning, thorough alternate rinsing in hot and cold water and finally a suitable acid dip.

Actual plating is effected either by procedures generally employed for other metals or by slight modifications of such procedures.

LACQUERING AND ENAMELING:

As is the case with plating, quality and adherence of organic coatings depend in great measure on the thoroughness with which the parts are cleaned prior to application of the finish. Cleaning procedures used on parts to be enameled or lacquered are similar to those used where plating is required. Most zinc-alloy die castings which are finished with lacquers or enamels, are phosphate pretreated to improve adhesion. Organic coatings commonly applied to metals fall into two general classes: Those which harden at room temperature and those requiring baking. Usually the baking-type finishes are more strongly adherent to zinc than the air-drying type.

RESISTANCE TO CORROSION

All three of these alloys offer resistance to atmospheric corrosion approximately on a par with rolled zinc and galvanized iron. Where close fitting parts are exposed to the weather and must be kept easily movable, the thin film of corrosion products which may cause binding can be avoided by using the Cronak Process[1] which forms a thin golden-brown corrosion-inhibiting film on the part surfaces. In general, no lubricants of animal fat origin ever should be used with parts made of these alloys. Data on the corrosion resistance of these alloys to specific materials follows:

Hydrocarbon Fuels and Lubricants: In the absence of moisture, zinc-alloy die castings are strongly resistant to attack by acid-free hydrocarbons. In the presence of water some corrosion takes place which, while not seriously detrimental to the strength of the casting may, in the case of fuel-handling devices, create some binding or clogging effect. While the best means of preventing such corrosion is to keep water out of fuel that comes into contact with the die castings, many cases exist where this cannot be accomplished economically. In a number of such instances use of the Cronak treatment is standard practice.

Ink: As the use of zinc engravings and lithographing plates in the printing industry indicates, printing inks have little or no effect on zinc. However, zinc-alloy die castings are strongly attacked by ordinary writing ink and are not recommended where this is present.

Alcohol: Although pure ethyl and methyl alcohol are considerably less corrosive to zinc than water, mixtures of alcohol and water are more corrosive than water alone. This fact, combined with the probable presence of iron rust, makes undesirable the use of zinc-alloy die castings in automobile cooling systems except in the presence of a suitable inhibitor. The use of zinc-alloy die castings in direct contact with potable alcoholic mixtures is not recommended.

Glycerine: While it is true that pure glycerine produces a smooth light etch on the surface of these alloys, glycerine-operated door checks and similar devices can be made satisfactorily of zinc-alloy die castings. Glycerine-alcohol mixtures produce only a light surface etching when a pure grade of glycerine is used. However, the presence of water or the use of a low-grade glycerine will result in pitting.

Soaps and Creams: Ordinary good grade laundry soaps have a definite inhibiting effect on the corrosion of zinc in hot water. Tests on die-cast zinc-alloy washing machine agitators, extending over several hundred hours of operation, clearly show the strong resistance of these alloys to corrosion in hot, soapy water. While nickel electrodeposits may be used as a means of improving the appearance of such parts, they are not required to improve serviceability. Cadmium coatings are a source of black discoloration on the work and should be avoided. The thin film of zinc stearate which forms on the surface of zinc in contact with soapy water has not only protective value but lubricating qualities as well.

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[1]U. S. Patent No. 2,035,380

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PROFESSIONAL VIEWPOINTS

"... jet propulsion made easy"

To the Editor:

I cannot refrain from commenting on the article on the jet propulsion engine in the July issue. As an engineer one may have no intention of working with such equipment and development; however, again as an engineer one is interested in something more reliable and technical in the way of discussion of new developments than is offered in the newspapers and "popular" periodicals. In his article Mr. Carmichael has done an excellent job of furnishing such a discussion. The basic fundamental theory is explained so simply as to be easily understandable. It is not complicated by minor details and variations. The whole general explanation is equally simple, permitting instructive and entertaining reading. Please continue your presentation of articles of this type.

—R. E. Orton, *Chief Engineer*
Acme Steel Co.

"... designing for adhesives"

To the Editor:

We have read with interest John Delmonte's article in the May issue of Machine Design entitled "Cemented Assemblies—Their Place in Design".

It has been our contention throughout, based upon our experience in manufacturing structurally with Cycle-weld cement, that in order to obtain 100 per cent efficiency with any adhesive in such manufacture, the place to begin is on the design board. Obviously, an assembly designed to be attached with rivets, spotwelds, bolts and screws, or other methods, will not allow the manufacturer to obtain 100 per cent efficiency from a tooling, labor or practical angle with adhesives.

Therefore, we congratulate Mr. Delmonte for his viewpoint, and also your magazine for its foresight and consideration of this most important matter—designing for adhesives.

—S. G. Saunders
Cycle-Weld Division
Chrysler Corp.

"... why use calculus"

To the Editor:

The article "Calculating Weights of Fillet Sections", published in the June issue of Machine Design, is interesting and potentially helpful if the calculator happens to have a working knowledge of calculus. It seems reasonable to suggest that, whenever possible, equations for such calculations should be derived from the simplest branches of mathematics. In other words, if the same result may be obtained by the use of geometry and trigonometry, why use calculus?

Geometric equations for computing the volume or weight of circular fillets are fairly simple for fillets having a 90-degree arc, and not too complicated for fillets with arcs of less than 90 degrees. Fundamentally, the calculations are based on the theorem of Pappus, which may be expressed as follows: Let A represent the area of a plane figure revolving about an axis in its plane but not cutting it, and let S represent the length of the circular arc traced by its center of gravity. The volume is equal to AS and, for one complete revolution, the volume will be $2\pi\rho A$, where ρ is the radius distance from the axis to the center of gravity of A.

In *Fig. 1a* the volume of the concave filleted section will be found by subtracting the ring generated by the sector ABC from the disk with radius R and thickness r. Any handbook will show the center of gravity of a 90-degree sector to be on a line parallel to one of the straight sides of the sector, and at a distance of $G = 4r/3\pi = .4244r$. The radius of the path described by the center

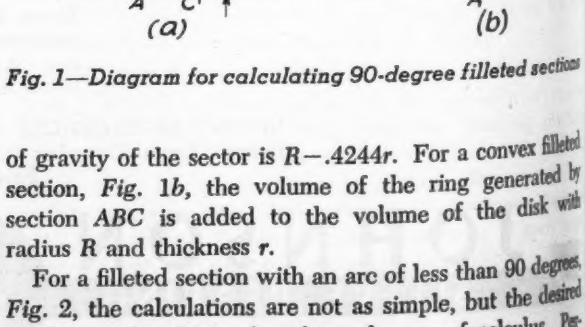


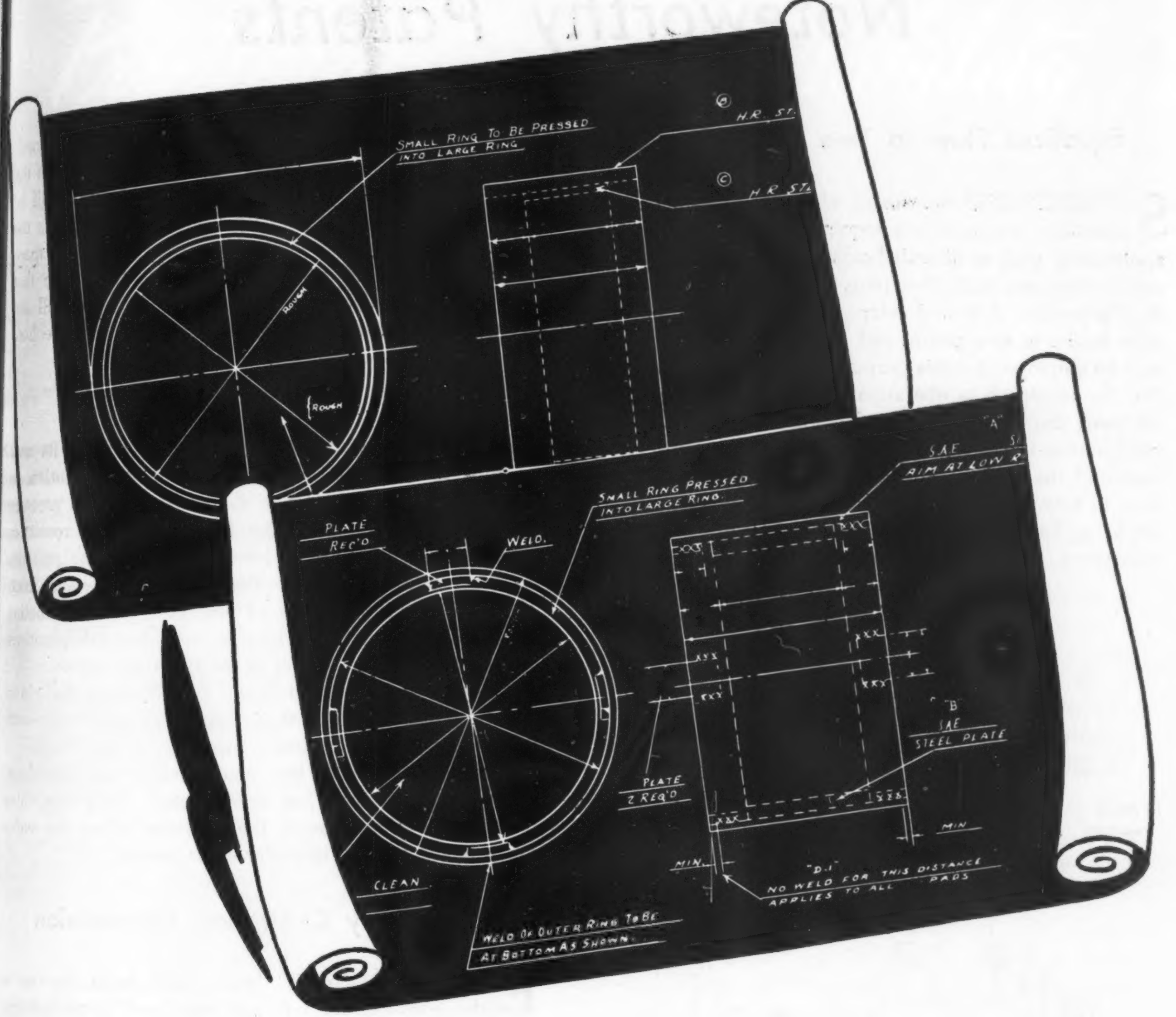
Fig. 1—Diagram for calculating 90-degree filleted sections

of gravity of the sector is $R - .4244r$. For a convex filleted section, *Fig. 1b*, the volume of the ring generated by section ABC is added to the volume of the disk with radius R and thickness r.

For a filleted section with an arc of less than 90 degrees, *Fig. 2*, the calculations are not as simple, but the desired result may be obtained without the use of calculus. Per-

(*Concluded on Page* 182)

Multiple piece motor frames—



TO YOUR SPECIFICATIONS

● Why not consult us about multiple piece motor frames and their advantages? Our experience covers a wide field of this type of work, including the forming and welding of the frame itself . . . shrinking an outer band over the inner band . . . welding of bracket pads and other multiple piece work for motors. We'll be glad to consult with you on the saving of materials and machining. We are prepared now to talk about this type of contract work for postwar production. Get in touch with us today.

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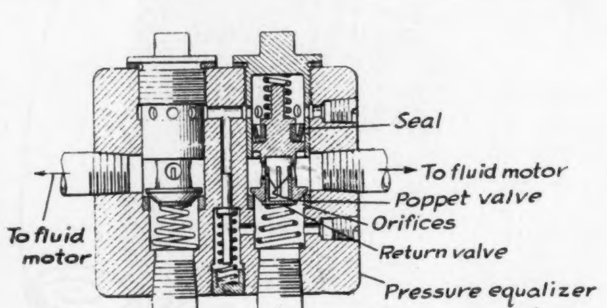
Noteworthy Patents

Equalizes Flow to Two Fluid Motors

SYNCHRONIZED movement of two fluid motors or hydraulic actuators is a requirement in a number of applications such as aircraft landing flaps, wing tip floats, and landing gear and other parts which must be actuated simultaneously. A flow divider consisting of two coupled units similar to gear pumps and actuated by the fluid flow may be employed for this purpose. However, in the event that the resistance to operation of the fluid motors is not the same due to differences in aerodynamic forces, etc., synchronism is not obtained because the volumetric efficiency of the gear type of divider varies with the resistance to flow. Means for imposing equal pressures on the discharge lines from the flow divider, thus insuring equal volumetric efficiency and equal flow, are covered by



Pressure equalizer insures equal discharge pressures from the two sides of flow divider, resulting in equal volumetric efficiencies and equal flow to the two fluid motors

patent 2,343,912 recently assigned to Pesco Products Co.

The complete equalizer assembly shown in the accompanying illustration consists of the flow divider and pressure equalizer. Pressure equalizer housing contains two identical valve assemblies each of which consists of a spring-loaded poppet-type valve, the upper part of which is in the form of a piston with a hydraulic seal. Spaces above the two pistons are interconnected.

Should the pressure of the fluid passing the right-hand valve increase over that passing the left-hand due to increased resistance at the unit actuated by the corresponding fluid motor, fluid will pass around the clearance of the right-hand piston and through the seal which, it will be noted, permits fluid to pass up through it but not in the reverse direction. This fluid will pass over into the space above the left-hand piston and will tend to force that piston downward inasmuch as the piston seal will not permit fluid to flow past it in the downward direction.

Pressure Difference Controls Valves

Pressure forcing the left-hand valve toward its seat is equal to the difference in pressures at the outlets to the two fluid motors. The increased drop in pressure across the left-hand valve due to the restriction results in creating a pressure at the left-hand inlet to the pressure equalizer exactly equal to that at the right-hand inlet. Thus the respective sides of the flow divider function with equal volumetric efficiency and identical quantities of fluid will be transmitted to the two fluid motors.

To effect the return of liquid forced above the piston there is provided a central passage in communication with the spaces above the piston and connecting through a nonreturn valve with the space below the right-hand valve. Liquid which has accumulated above the valve thus leaks back whenever the pressure below the valve drops below the pressure above the piston.

Electrically Controlled Transmission

ELECTROMAGNETIC brake and clutch devices of the eddy-current type are employed in an automotive transmission design covered by patent 2,343,291, recently assigned to the Chrysler Corp. The complete transmission includes a manually controlled two-speed and reverse synchromesh unit and a governor-controlled two-speed planetary unit. For the lower of the two speeds in the planetary unit the sun gear is held against rotation by means of the eddy-current brake, while the engine drives the ring gear and the output is taken from the spider which carries the planet pinions. This gives speed reduction and torque multiplication as required for starting and acceleration. One-to-one ratio is obtained by de-energizing the brake and energizing the eddy-current clutch which locks the planetary gearset.

Applied to an automotive transmission, the planetary unit functions automatically at predetermined engine speeds to change the ratio between first and second speeds, or between third and fourth, depending upon the ratio of the manually controlled unit selected by the driver. The shift between second and third is effected manually, the controls providing for momentary de-energizing of both brake and clutch to allow the sliding shift to occur without the necessity of pedal de-clutching.

ASSETS to a BOOKCASE

Centrifugal Pumps and Blowers

By Austin H. Church, associate professor of machine design, New York University; published by John Wiley & Sons, Inc., New York; 308 pages, 5⅝ by 8⅞ inches, clothbound; available through MACHINE DESIGN, *$4.50 postpaid.*

Whether your interest is in actually designing centrifugal pumps and blowers, or in engineering such commercial units into some kind of mechanical equipment, this book will give you the required data. As the author points out in his preface, the application of these units is well-nigh universal, many fields would be seriously handicapped without them, and it is surprising that there are so few modern texts on the subject.

The book is at once basic and technical, which sounds paradoxical but is nevertheless true. This is just another way of saying that the author takes care to explain things in everyday English before expounding in the language of higher mathematics. After chapters dealing with principles of fluid flow and basic theory of design, the author proceeds to sections on speed and efficiency, design of vanes, pump impeller types, details and materials, applications and selection, etc. It is an authentic little volume, adequately illustrated.

□ □ □

Process Practices in the Aircraft Industry

By Frank D. Klein Jr., senior metallurgist, Materials Control branch, United States Army Air Forces; published by McGraw-Hill Book Co. Inc., New York; 266 pages, 6 by 9 inches, clothbound; available through MACHINE DESIGN, *$2.75 postpaid.*

Working in collaboration with numerous materials, parts, and aircraft producers, as well as with many authorities on processing, the author has prepared a good practical reference book. It generally is recognized that the most significant recent advances in engineering methods have been made in the aircraft industries and many engineers working in other fields can learn much by studying aircraft-industry procedures. In this book are discussed the latest methods utilized in processing materials and parts for use in aircraft.

Procedures followed in procurement are dealt with at the outset, accompanied by extensive tables of Army and Navy specifications applying to most of the materials used in aircraft. Aluminum and magnesium are covered at considerable length, but the data on steels seem abbreviated. Some fine tips on design and drafting technique are presented with the aid of neatly rendered drawings and there are a wealth of data on corrosion-resistant and decorative finishes.

Practical Design for Arc Welding

This is a refreshingly different kind of technical book for it tells its story in pictures and symbols rather than in words. Do you want to know how a machine base, a gearbox, a radiator shell, a fan rotor should be welded? A glance at the drawing of a similar unit along with its symbols in this book will show you the proper type of weld to specify.

We need more books like this one. Its author recognizes that the machine designer is not as greatly interested in lengthy dissertations on welding theory, metallurgical aspects, etc., as much as he is in purely practical information—the "end results" as it were—which he can put to good use in his everyday work.

If you have little time for study, yet are eager to keep abreast of modern arc-welding practice, this book is for you. It is authored by Robert E. Kinkead, consulting engineer on welding, is published by The Hobart Bros. Co., Troy, Ohio, and is the first volume of a three-volume set. It contains 100 design plates, 8¾ by 11½ inches, is clothbound and is available for $3.50.

The Oxy-Acetylene Handbook

There is little about oxyacetylene welding that one cannot find in this book. It seems to tell the whole story. For the serious student, there is a good deal about historical developments, metallurgical aspects, etc. For the practical man who is primarily concerned with "how to do" and "what to specify", the bulk of the book offers much on the oxyacetylene welding of common commercial metals, both ferrous and nonferrous.

It is a well-prepared manual, amply illustrated, contains 587 pages, 6⅛ by 9¼ inches, is published by The Linde Air Products Co. of New York and is available for $1.50.

Angular Spacing Tables

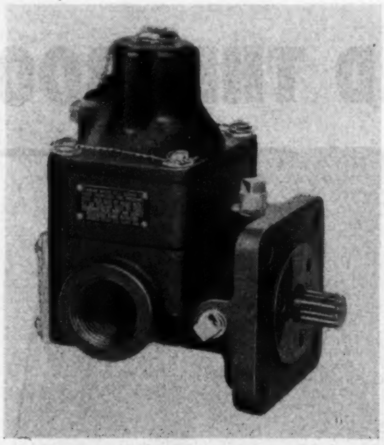
An elaborately prepared loose-leaf book of angular spacing tables has recently been compiled and published by Vinco Corp. Used for computing and checking the spacing of teeth in precision gears, splines, index plates, etc., the tables—covering all spacings up to 200—list divisions carried out to one-thousandth part of a second. The appendix of the book gives fundamental dividing angles in degrees, minutes and seconds, degrees and decimals of the degree and in circular measure (radians). Conversion tables and important constants are included also.

For the design and checking department of concerns engaged in the manufacture of precision gears, splines, etc., this book should prove extremely helpful. Carrying the title "Angular Spacing Tables", its price is $10.00.

New PARTS AND MATERIALS

Engine-Driven Fuel Pump

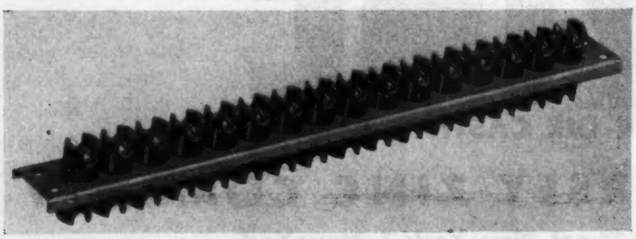
ORIGINALLY DESIGNED for aircraft but adaptable to other applications for liquid handling, a new lightweight, high-pressure engine-driven pump has been announced by Romec Pump Co., 101 Abbey road, Elyria, O. This gasoline pump weighs 2¾ pounds and delivers 400 gallons per



hour at 2500 revolutions per minute. Its pressure range is from 6 to 35 pounds per square inch. Designed to withstand extreme temperatures, the pump is precision-built with balanced type relief valves and with low-temperature shaft seals.

Terminal Block for Subpanels

FOR ELECTRONIC and electrical designs requiring external terminals, the new feed-through terminal block of Curtis Development & Mfg. Co., 1 North Crawford avenue, Chicago, is proving convenient and simple for subpanel mounting. The terminal block consists of individual feed-through terminals, molded in bakelite and permanently held in a metal strip in any combination desired. Production now includes blocks having any number of units from one to sixteen. However, because of the sectional design, blocks can be supplied with any number of terminals desired. Ample clearances and leakage



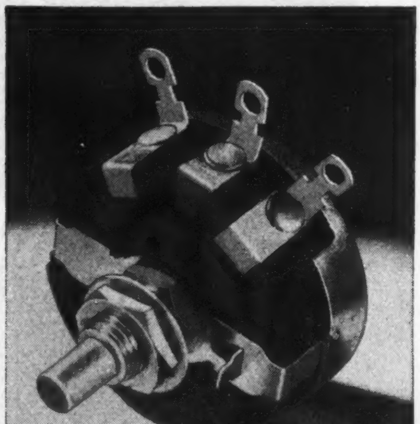
distances are provided for circuits carrying up to 300 volts, 20 amperes. Center to center distance between terminal units is ⅝-inch. No. 8 screws are used on each side.

Phenolic Molding Compound

MICA-FILLED and based on a newly developed phenol formaldehyde, a new material designated as Resinox 7934 has been announced by Monsanto Chemical Co., St. Louis 4. Its principal advantages as an insulating medium in high frequency apparatus are low dielectric constant and power factor, low water and moisture absorption, and relatively high heat resistance.

Potentiometer Redesigned

CAPABLE OF withstanding extreme vibration and mechanical abuse encountered in wartime service, a new version of the Type 58 wire-wound potentiometer or rheostat has been developed by Clarostat Mfg. Co. Inc., 285-7

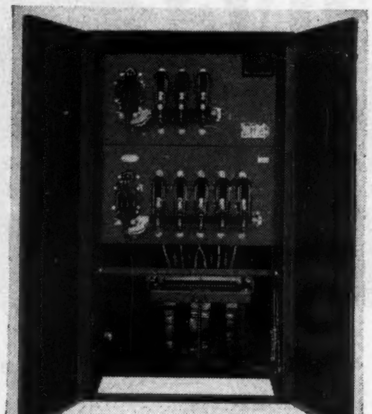


North Sixth street, Brooklyn. This new control differs somewhat from the previous Type 58. A metal strap on the shaft face provides for the two-position locating pin which cannot break or tear off. Also, the metal strap grounds the metal cover which is keyed in place on the casing. The bushing being keyed into the bakelite case cannot slip or turn when the locking nut is drawn up tightly. Corrosion and electrolytic action, especially when the control is used on direct current, is eliminated by the use of bakelite. The center rail and terminal comprise one piece, and a direct connection between wind-

ing and the "L" and "R" terminal lugs is provided. Available in tandem units with two or more controls on a common shaft, the control has a 1500-volt breakdown insulation between winding and shaft. Ratings are: Linear, 3 watts; V and W tapers, 2 watts; L, N and U tapers, 1.5 watts. Resistance values: Linear, 1-75,000 ohms; tapered, 10 to 50,000 ohms.

Reduced Voltage Motor Starter

CALLED TYPE "ARC" a new magnetic, reduced voltage motor starter protects motors from sustained overloads, locked rotor condition, single-phasing and overloading from too frequent starting. Built by Allis-Chalmers Mfg. Co., 1126 South Seventieth street, Milwaukee, the starter is now available for control of motors from 5 to



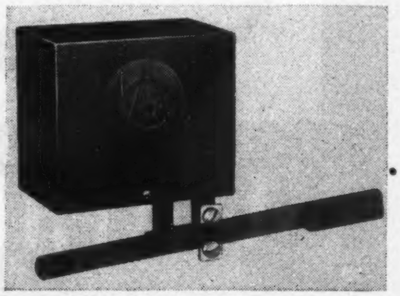
2500 horsepower, up to 5000 volts, 3-phase, 50 or 60 cycles. Reduced voltage, two-point starting is obtained with a built-in autotransformer, utilizing accurate, synchronous motor-driven, adjustable timing relay for transition from starting to running position. The autotransformers are built into units up to 800 horsepower; above 800 horsepower autotransformers are furnished for separate mounting. Interrupting capacity rating of the new starter is ten times the motor's full load current.

Electrode for A-C Operation

MADE SPECIFICALLY for alternating-current operation, a new general-purpose electrode for mild steel, designated as "Fleetweld 35", has been announced by The Lincoln Electric Co., Cleveland. The new electrode will also operate on direct-current with either polarity, depending upon the type of work being done, and has characteristics similar to the company's "Fleetweld" electrodes for high-speed welding of single or multiple passes for work in flat, vertical or overhead positions. Available in ⅛-inch, 5/32-inch and 3/16-inch sizes, the electrode provides a tensile strength from 62,000 to 70,000 pounds per square inch in the as-welded specimens; yield strength of 52,000 to 57,000 pounds per square inch, with ductility (elongation in 2 inches) of 23 to 30 per cent. Made in 14-inch lengths the electrode conforms to American Welding Society specifications for arc welding electrodes classes E-6010 and E-6011.

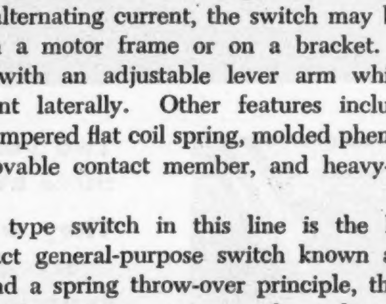
Switches with Spring Action

TYPE L-100 single-pole float switch is the first of a new line of inexpensive switches being produced by Automatic Control Co., 1005 University avenue, St. Paul 4. It can be used for small pumps and motors of industrial and



domestic equipment where durability and positive action are required. Rated at 2 horsepower at 110 volts and 220 volts alternating current, the switch may be mounted directly on a motor frame or on a bracket. It also is equipped with an adjustable lever arm which can be readily bent laterally. Other features included are a specially tempered flat coil spring, molded phenolic mounting for movable contact member, and heavy-duty silver contacts.

Another type switch in this line is the heavy-duty, silver-contact general-purpose switch known as Type L. Built around a spring throw-over principle, the switch is provided with a wiping action on the make, maintaining



constant pressure on the heavy silver contacts. Action of the spring is such that the pressure is increased before the fast break, eliminating cause for breakdowns. This switch is also rated at 2 horsepower at 110 and 220 volts a. c., and at ½-horsepower at 110 and 220 volts d. c.

Synthetic Rubber for Insulation

SYNTHETIC RUBBER latex insulation for power, lighting and communication cable is a result of wartime developments in rubber, according to United States Rubber Co., New York, who developed the new Nubun insulation material. It will permit the design of new types

. . . and a few often-neglected factors that affect it

The *thermal conductivity* of a metal is one thing.

The actual *heat transfer* from one fluid to another through the metal may be a different story altogether.

For thermal conductivity is listed on an *ideal* working basis. It does not take into account the practical factors that may slow down heat transfer under working conditions.

It does not allow, for instance, for corrosion deposits and films that form on metallic walls or for heating or cooling gases that cling to the metallic walls. These films have such low conductivity that they often make relatively unimportant the conductivity of the metal itself.

The diagram (below) illustrates the slow-up in heat transfer from corrosion products and film. The table lists the conductance ranges for metal walls, and for films that may form upon them.

These comparative figures show why other factors than rated thermal conductivity are of greater importance in heat transfer equipment.

All the INCO Nickel Alloys are highly resistant to corrosion, even at high temperatures. They develop very thin and tenacious oxides within their limiting temperature range. They do not scale or build up heavy insulating jackets of corrosion deposits. Their clean smooth surfaces keep down accumulations of sludge and deposits which might retard heat transfer.

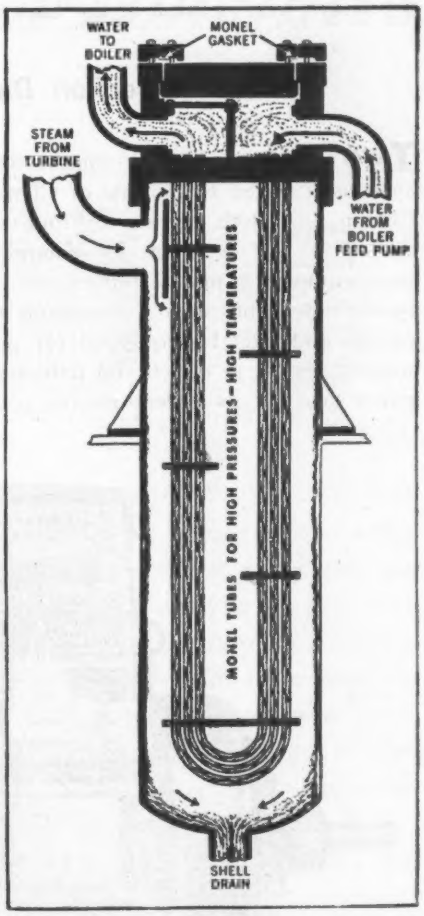
These are the reasons for the remarkably rapid heat transfer rates which nickel and the INCO Nickel Alloys provide under actual service, especially where corrosive conditions and high temperatures are encountered.

* * *

A practical example which illustrates the usefulness of INCO Nickel Alloys for heat transfer work is the performance of Monel tubing used in three of the latest high-pressure power stations:

TWIN BRANCH PLANT, of the Indiana and Michigan Electric Company, a subsidiary of the American Gas and Electric Service Corporation. The plant was designed to generate steam at a new high pressure for commercial units . . . 2500 lbs. Steam bled from the turbines at temperatures as high as 637° F., and pressures as high as 700 psi, surrounds the Monel tubing through which 550,000 pounds of water an hour rush under pressure of 3200 psi. Despite the speed at which it moves, the water is boosted from 353° F., to 490° F., due to the excellent heat transfer properties of the Monel tubing.

NARRAGANSETT ELECTRIC CO., Providence, R.I. Steam is bled from the turbine at 700° F. 430,000 pounds of water, under pressure of 1300 psi, pass through the Monel tubing every hour. The water is pre-heated to 400° F., before entering the high-pressure boiler.



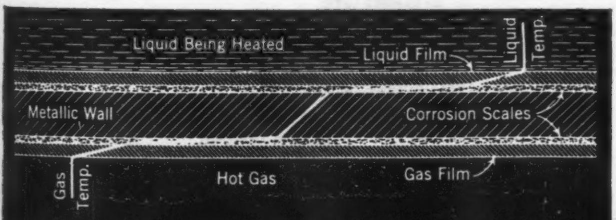
Cross-section of typical heat exchanger of the type used in Narragansett Electric Co. Plant. Note use of Monel Tubing.

OSWEGO PLANT of the Central New York Power Corp. Steam is bled from the turbine at 770° F., at a pressure of 600 psi. Internal water pressure in the Monel tubing is 2000 psi. The swiftly rushing water is pre-heated to 450° to 485° F.

Recently, all three installations were inspected. In addition to performing their heat transfer job, the Monel tubes in these heaters have stood up perfectly under the high pressures, high temperatures, and corrosive, erosive attack to which they have been subjected.

For further information about heat transfer, write for the INCO Technical Bulletin: "Heat Transfer through Metallic Walls." Address:

THE INTERNATIONAL NICKEL COMPANY, INC.
67 WALL STREET NEW YORK 5, N. Y.



A section through a metallic wall transferring heat from a hot gas to a cooler liquid, showing gas and liquid films, corrosion deposits, and the general effect of each upon the temperature gradient.

Comparison of the Conductance Ranges for Metallic Walls and Films

Nature of Obstruction	Conductance Range Btu./hr./sq. ft./°F
Metal wall	
1 in. thick	75 to 3,000
0.025 in. thick	3,000 to 120,000
Condensing vapor	1,000 to 4,000
Evaporating liquid	500 to 2,000
Liquid being heated	50 to 500
Liquid being cooled	40 to 400
Gas being heated	5 to 50
Gas being cooled	5 to 40
Corrosion scale	5 to 20

of wire and cable with improved electrical and physical characteristics. Made by the latex continuous dip method, Nubun insulation has several outstanding features—flexibility, impermeability to water, laminated construction, and centering of the conductor to produce an insulated wire of maximum conductivity and minimum diameter. It is said to be homogeneous following vulcanization and has high electrical characteristics such as dielectric strength and insulation resistance. Made from a special modification of Buna S synthetic rubber, it is low in specific conductive capacity, has good aging qualities, and will resist severe wear because of the latex process.

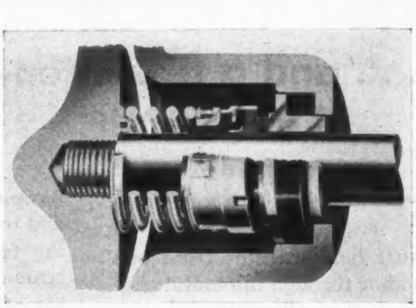
Solid Injection Diesels

TWO SOLID injection, compression ignition engines have been added to the line of Climax Engineering Co., Clinton, Ia. Both engines are four-cycle, full diesels intended for use as drives for pumps, compressors, mills, mine equipment, marine service, etc. Model D 148 is a two-cylinder unit with a maximum rating of 22 horsepower, and may be equipped for pulley drive with or without clutch, or clutch and reduction gear, or auxiliary power takeoff. As a diesel electric plant it may be direct-

connected, on a single base, with a 15-kva generator. The other engine, Model D 297, is a four-cylinder diesel with a maximum rating of 44 horsepower. Drive equipment similar to Model D 148 is available. A special feature of this model, however, is that a flywheel, clutch, generator or marine gears may be installed on either or both ends, providing a radiator is not used.

Bellows Type Shaft Seal

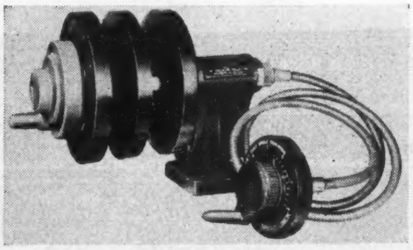
SINCE PREPARING the announcement on the bellows type shaft seal developed by The Crane Packing Co., 1825 Cuyler avenue, Chicago 13, which appeared in the August issue of MACHINE DESIGN, additional data has become available. The normal range of operating pressure is from 0 to 200 pounds per square inch, although the seal has been applied successfully to much higher pressures. It is regularly available for shafts from ⅝-inch to 3 inches in diameter, and larger sizes to 10 inches in shaft diameter can be furnished on application. An exclusive feature of the seal is the flexing head of the bellows which offers no resistance to the spring, allowing the sealing washer to remain in contact with the seat as the sealing faces wear, and providing automatic com-



pensation for shaft vibration and end-play. A protecting ferrule, fitting the shaft loosely, guides the flexing bellows head and prevents the synthetic rubber from contacting the shaft at this point. The stationary floating seat is held in a synthetic rubber holding ring, permitting easy insertion, preventing stress distortion of the sealing face during installation, and allowing the face to be lapped before insertion.

Variable-Speed Transmission

ADAPTABLE TO a variety of machines such as band saws, bottle capping machines, conveyors, embossing machines, etching machines, machine tools, mixers, ovens, printing and binding machinery, etc., a new variable-speed transmission added to the line of American Pulley Co., 4200 Wissahickon avenue, Philadelphia 29, features simplified remote control and universal mounting. Re-



mote control through a compact flexible shaft makes possible the mounting of the unit in or on machines to place the control in a safe and handy location. Because the movement of the flanges is controlled mechanically, the Speed-Jack transmission may be mounted vertically, horizontally or in any other position suited to the equipment. V-belts provide stepless control of speed through a 3 to 1 ratio. The design of the control mechanism eliminates ratio-creep and assures belt alignment regardless of the

4 Ways Needle Bearings Promote Cost Economies

TYPE NCS *TYPE AT* *TYPE DC* *TYPE FDT* *TYPE PN*

LOWEST COST PER GIVEN LOAD OF ANY ANTI-FRICTION BEARING IS BUT ONE OF THE NEEDLE BEARING'S CONTRIBUTIONS TO COST SAVINGS

•

One of the surprising features of Torrington Needle Bearings, particularly in the case of the widely used D.C. Type, is their low cost—for they are recognized as the lowest in cost per given load of any anti-friction bearing!

Yet initial cost is only one of several economy features which accrue to users of these modern anti-friction units. Consider, for example, the advantage of their high unit capacity in relation to cost. The smaller *size* of the Torrington Needle Bearings which may be used results in savings of size, weight and material costs of surrounding members.

This contributes in turn to easier, faster handling of all parts on the production line. But it is the ease and speed with which Torrington Needle Bearings are installed that effect a major saving in time and labor required for assembly. Furthermore, neither the use of snap rings nor staking the bearings in position is required—as once properly installed Needle Bearings will not shift or creep in the housing.

Efficient lubrication—due to the basic design of these Needle Bearings—eliminates the need for complicated or costly *systems* for providing ample lubrication. This means simplification of the design and production jobs.

Economies to Aid User

All of these advantages add up to still another important cost consideration—a longer life of trouble-free service—for the product using Torrington Needle Bearings. Reduced costs of servicing, replacement parts and repair, along with lower operating costs due to increased product efficiency are features your *customers* will also appreciate.

If you are seeking manufacturing economies and wish to increase your product's efficiency at the same time, you may find the answer in Torrington Needle Bearings. Our Engineering Department will gladly work with you in securing the full advantages of these bearings in *your* own product planning. Write for further information. A copy of our Needle Bearing Catalog No. 30-A should be in every product engineer's file. Send today for your copy.

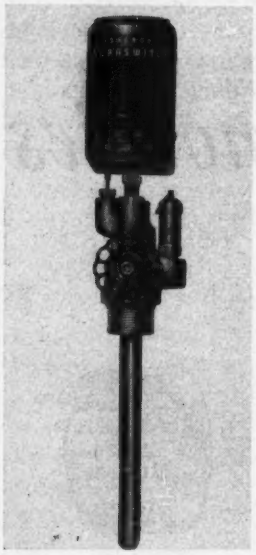
THE TORRINGTON COMPANY
Established 1866 • Torrington, Conn. • South Bend 21, Ind.
"Makers of Needle Bearings and Needle Bearing Rollers"

New York, Boston, Philadelphia
Detroit, Cleveland, Seattle
San Francisco, Chicago, Los Angeles
Toronto, London, England

TORRINGTON
NEEDLE BEARINGS

mounting position, belt wear or stretch. The flanges are of steel-faced plastic which results in light weight and provides balance and smooth, vibration-free operation.

Compressor Unit Available



COMBINING SEVEN devices for the control of compressor operations into a compact unit, the new Compressor-Trol developed by the Electro-Mechanical division of Manning, Maxwell & Moore Inc., Bridgeport, Conn., is available in three sizes. These are ½, ¾ and 1¼ inches, depending on the inlet connection. The control is equipped with the company's Duraswitch, with either an electric or mechanical attachment for operating the two-way unloading valve mounted on the Compressor-Trol casting. All requirements of tank-mounted compressors up to 15 horsepower or 60 cubic feet per minute are covered in the new unit. The muffler on the compressor discharges into the tank, breaks up the discharge and disperses the air in a manner insuring adequate mixing with the cooler gases in the tank to reduce the temperature and moisture content. The combined Duraswitch and gage has a heavy-duty gearless movement, slide rule dial, a horizontal self-draining and nonfreezing tube, and the entire gage is removable. It can be furnished in all standard pressure ranges.

Arc-Welding Electrodes

SMOOTH-FLOWING, arc-welding electrodes with quiet arc characteristics have been introduced by Allis-Chalmers Mfg. Co., Milwaukee. These new electrodes, both alternating and direct-current types, use the standard American Welding Society number and color, simplifying selection and handling.

Retractable Tubing

TO MEET THE many industrial requirements for a portable duct connection for cold or heated air, The Wiremold Co., Hartford 10, Conn., is offering its new specialized tubing. Made of lightweight fabric, a 15-foot length when retracted may be stowed in a 1-foot container, although when of heavier material this ratio is somewhat reduced. The standard inside diameters include 4, 6, 8, 10 and 12 inches, while other diameters are made to specifications. The lengths in which it is available are from 2 to 50 feet, and temperature of heated air it is capable of withstanding is up to 300 degrees Fahr. End terminations are furnished to suit individual requirements. Many variations in construction are possible depending on the application. The number of spirals may be increased or decreased from the standard which is four per foot length. Also many types of flexible material can be used, ranging from lightweight cotton or synthetic fabrics to heavy canvas duck, which may be impregnated where desirable. Some of the typical applications include airplane engine preheaters, ventilating ship holds, tanks, cars and other enclosed spaces.

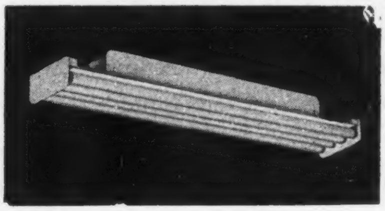
Steel Aircraft Wheels

TO MEET THE specifications and requirements of a conventional light-metal wheel, such as aluminum, magnesium, etc., The Kelsey-Hayes Wheel Co., Detroit, has developed a steel aircraft wheel. The new wheel is made from steel stampings of low-alloy material, heat treated, and is designed to provide the maximum of strength with minimum weight. Riveted type of construction is employed.

Engineering Dept. Equipment

Lights for Drafting Room

BOTH THE pendant type (No. 1946-C) and the ceiling-mounted type (No. 1947-C) luminaires, announced by Curtis Lighting Inc., 6135 West Sixty-fifth street, Chicago 38, can be used both individually and in continuous rows in drafting rooms. The luminaires are unshielded units and can be converted into "Warrior" luminaires by obtaining the louver body and louver end plates. A steel



wiring channel and a nonmetallic reflector, ballast shields and reflector ends are used. The width of the unit is 9½ inches, and length 48 inches. Suspension from the top of the ballast to the bottom of the wireway end plate is 6⅛ inches. The unit is wired for 110-125 volts, 60 cycles, alternating current, high power factor tulamp ballasts. It can also be furnished wired for 220-250 volts.

Blue, Black and White Printer

"OVER-ALL DRYING" of blue, black or white prints is performed by an electrically heated dryer offered by Peck and Harvey, 4327-M Addison street, Chicago 41. Available in two sizes, the machine can handle prints of 26 and 44-inch widths. In addition to the standard heat regulation, the "B-8" dryers are also equipped with variable-speed drive motors and controllers which permit

Preview of Dependability

American Flexible Metal Hose Proves Its Fitness Before Leaving The Factory

BUY WAR BONDS
Buy More Than Before

EVERY LENGTH of American Flexible Metal Hose, before leaving the factory, is thoroughly tested under pressure as a final check. This operation is typical of the care exercised in the manufacture of AMERICAN products, and is one of the reasons why AMERICAN maintains such an enviable reputation for dependability.

Wherever a practically indestructible, flexible, airtight conveyor is required for oil, gas or liquids; for vacuum service or for isolating vibration, it is logical to turn to American Flexible Metal Hose.

American Flexible Metal Hose withstands heat and high pressures without deterioration. It is resistant to many of the corrosive influences which render non-metallic hose impractical to use. Furnished in bronze and other workable metals with or without couplings, there is a type for nearly every industrial purpose.

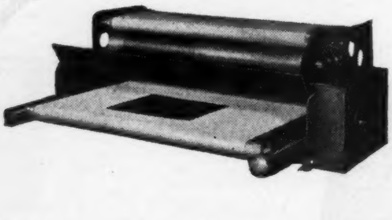
The Technical Department of The American Metal Hose Branch of The American Brass Company is prepared to offer suggestions and make recommendations regarding special applications. 44100

ANACONDA

American Metal Hose

AMERICAN METAL HOSE BRANCH OF THE AMERICAN BRASS COMPANY • General Offices: Waterbury 88, Conn.
Subsidiary of Anaconda Copper Mining Company • *In Canada:* ANACONDA AMERICAN BRASS LTD., *New Toronto, Ontario*

speed changes over a range of 6 inches to 3½ feet per minute—thus accommodating various types of work under different drying conditions. The current consumption on the 26-inch dryer is 14 amperes on 110 volts, and 7 amperes on 220 volts; and on the 44-inch unit, 23 amperes on 110 volts, and 12 on 220 volts. The latter model, furnished for alternating-current or direct-current operation, is 58 inches long, 28 inches wide and 13 inches high. The overall dimensions of the 26-inch dryer, also for alternating-current and direct-current operation, are 40 x 28 x 13 inches.



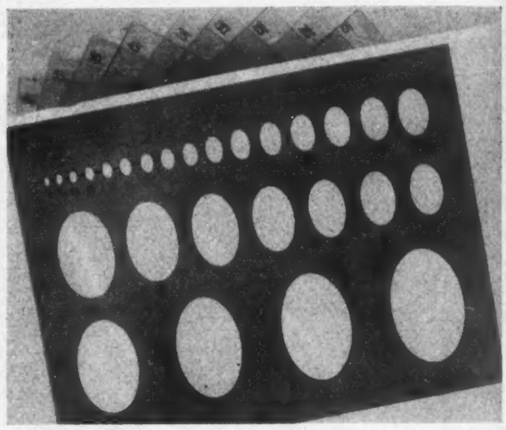
Microfilming Drawings

FOR CONDENSING and preserving vital records for an indefinite period, the Microcopy Corp., 2800 West Olive avenue, Burbank, Calif., has introduced its Translite Hi-Reduction process. Because of this high-fidelity translite feature, the process is particularly adaptable to engineering drawings in pencil on transparent paper. The negative can be used to make copies on other pieces of film, and to enlarge and make copies of the original in any size desired, on tracing paper, cloth or film. The

Microcopy reduction photographs of drawings are on a continuous roll, making the master tamper-proof as well as saving space in storage. For reading copies, a multiple-magnification viewer is supplied in various models which is operated on a motion-picture film projector principle, from a control panel. This facilitates finding locations on films by permitting enlargements to be made on a screen, as well as permitting a portion of a drawing to be centralized within the frame for easier reading.

Three-Dimensional Drawing Guides

THREE-DIMENSIONAL drawing is speeded up by the ellipse templates now being offered by The A. Lietz Co., 913 South Grand avenue, Los Angeles 15, and 520 Montgomery street, San Francisco 11. These guides are .02-thick celluloid templates and are made in a set of ten, with angles from 15 to 60 degrees by 5-degree increments.



Templates of 60, 55, 45 and 40 have ellipses from ⅛-inch to ⅜-inch, increasing by thirty-secondths, then by sixteenths to 1 inch and by eighths to 2. Templates of 35, 30, 25, 20, and 15 degrees progress the same, but start with sizes of 5/32, 3/16, 7/32, ¼ and ⅜-inch, respectively. The guides are equally efficient for either pencil or ink, and are available as complete sets or separately.

Fluorescent Unit Announced

ANNOUNCED BY F. W. Wakefield Brass Co., Vermilion, O., a fluorescent unit has been designed for high quality illumination in drafting rooms. It uses etched ribbed glass for side panels to prevent glare and open louvers in the bottom of the unit to prevent direct view of the lamp. Lamp replacement is provided by a hinging arrangement of the louvers whereby lamps can be easily reached. Known as the Beacon, the unit has an all-steel construction with the exception of the louvers. The metal end-caps are pierced in a decorative design, backed up with translucent plastic. Ballasts are mounted in an inverted position within the channel and are only partially enclosed. This construction provides a cooler operating position with resultant decreased failure of ballasts from heat exposure. Metal parts of the unit are finished in satin zinc, providing a rustproof plated coating. In addition one coat of clear enamel is applied for extra protection. Units are available with stem suspension or with close-up mounting for low ceiling areas.

Men of Machines

WIDELY KNOWN as an industrial engineer, Paul E. Frantz, in his new appointment as operations manager of Adel Precision Corp., brings to the company an extensive experience in the manufacture of precision armament equipment, and in the mass-production of electrical household appliances. Acting as a consulting engineer he has designed and tooled intricate war equipment such as the automatic pilot, K-9 gunsights, turbo-superchargers, 90-millimeter antiaircraft guns, naval gunfire directional controls and electrical components such as commutators used in inertia engine starters for aircraft. Prior to this, Mr. Frantz was chief engineer of the Apex Electrical Mfg. Co., Cleveland, manufacturers of domestic machinery, electric motors and related devices. He holds a number of patents on several such appliances. In addition to being a graduate electrical engineer, Mr. Frantz is a lawyer, a member of the Ohio State Bar, where he specialized in research on patent law. He is a graduate of Notre Dame. Before joining Adel, he had been associated with Lucien I. Yeomans Inc., industrial engineers of Chicago.

ELECTION OF P. H. Bates of the National Bureau of Standards, Washington, as president of the American Society for Testing Materials has been announced recently. Mr. Bates has been active in the society since 1926, and in 1940 was the society's Edgar Marburg Lecturer. From 1937 to 1939 he was also a member of the executive committee, and prior to his appointment as president he completed a term as vice president of the society. A graduate of the University of Pennsylvania in 1902, Mr. Bates' first connection was that of assistant chemist of the Pennsylvania railroad, where he served until 1906. While at Altoona, Mr. Bates was associated with the society's first president, Charles B. Dudley. After leaving the railroad, he became chemist of the Technologic Branch, U. S. Geological Survey at St. Louis, and from 1910 to 1919 was in charge of the Pittsburgh Branch of the National Bureau of Standards. During the next two years he was chief, Structural Miscellaneous Materials Division, at the Bureau, and also acting chief, Ceramic Division. He has been in his present position as chief of the Clay and Silicate Products division since 1921.

AFTER SERVING for four years as assistant chief engineer of the Oldsmobile division of General Motors Corp., Lansing, Mich., Jack F. Wolfram has been made chief engineer. Born in Pittsburgh, he obtained his early schooling there, and secured his technical training through courses in drafting, machine design, and business administration from several national correspondence schools, as well as private tutoring. Mr. Wolfram began his career as a technical clerk during the summers of 1916 and 1917 with the Hyle and Patterson Co., and later in the blueprint department of H. Koppers Co. The following nine months he spent as a reporter for the F. W. Dodge Co., and in 1918 joined the civil engineering staff of the B & O Railroad. Two years later he became connected with the American Heating and Ventilating Co., Cleveland, in charge of heating and ventilating design, and left there after six months to enter the tool designing department of the Enterprise Tool Co. From 1921 to 1928, he was employed by the Chandler Motor Car Co. successively

as a draftsman, designer and experimental engineer. In 1928 he entered the employ of General Motors as assistant experimental engineer of Oldsmobile, and in 1934 was advanced to the position of experimental engineer. He was named assistant chief engineer in 1940, which position he filled until his recent appointment as chief engineer. Mr. Wolfram is a member of the Society of Automotive Engineers.

♦

H. S. Manwaring has recently been named chief engineer of the mechanical research and development division of International Harvester Co., Chicago.

♦

Raymond G. Hilligoss has left his post as assistant professor of engineering for Oklahoma A & M college, Stillwater, Okla., for his new position as lubricating engineer with Boeing Aircraft Co., Wichita, Kans.

♦

John Seagren, formerly chief engineer of Atlas Imperial Diesel Engine Co., Oakland, Calif., is now connected with Northern Pump Co., Minneapolis.

♦

Charles I. Preston recently was appointed aircraft engine designer for Chrysler Corp., Detroit. He formerly had been connected with the U. S. Army Air Forces, Air Service Command, Patterson Field, Fairfield, O.

♦

Howard H. Langdon has been made head of research and development of Consolidated Machine Tool Corp., Rochester, N. Y. He formerly had been head of the mechanical engineering department, State College of Washington.

♦

Jack Brocken, who has been an associate designer with Brooks Stevens Industrial Design, has joined the firm of G. McStay Jackson Inc., Chicago, as vice president in charge of industrial design.

♦

John E. Tereschuk, previously associated with the White Motor Co., Cleveland, as automotive designer, has become connected with Chrysler Corp., Highland Park, Mich., in the same capacity.

♦

Eugene J. Freeman has been transferred from plant layout engineer to senior hydraulic design engineer of Adel Precision Products Corp.

♦

Werner H. E. Engel has been promoted from designer in charge of the drafting room to product engineer in the Photo-Record division of Remington Rand Inc., Brooklyn, N. Y.

♦

Joseph Norman Paquin has been advanced to the position of chief development engineer from development engineer at Weatherhead Co.

♦

Alex D. Bailey, vice president of the Commonwealth Edison Co., Chicago, is the nominee for president of the American Society of Mechanical Engineers.

♦

Walter P. Schmitter is the third recipient of the Edward P. Connell award, the presentation being made at the American Gear Manufacturers association at its recent annual meeting. Mr. Schmitter is chief engineer of The Falk Corp., and has been active in the association in many capacities, including that of president.

♦

A. E. Lindberg, associated with Moline Tool Co., Moline, Ill., for thirty-four years—the last twenty-five as chief engineer—has recently retired. Replacing him is C. E. Parkhurst who has been connected with the company for sixteen years. D. C. Eipper, who has been on the engineering staff for twenty-five years, has been appointed assistant chief engineer.

♦

Dr. L. H. Fuller has been appointed assistant chief engineer of the Joshua Hendy Iron Works, Sunnyvale, Calif.

♦

John A. Faler, formerly assistant chief engineer, has been appointed to take charge of extraction equipment development. Mr. Faler will coordinate new design and equipment development work in this division. Thomas J. Kearney has been promoted from technical advisor to the director of sales for Detrex Corp., to assistant chief engineer in charge of industrial equipment design and detailing. Mr. Kearney has been connected with the company for ten years.

♦

G. T. Mitchell, formerly production engineer, has been named superintendent of the machine shop of Ozalid Products division. A. C. Wedge is the production engineer. It was erroneously stated that Mr. Mitchell was made assistant to F. O. Trump, recently appointed chief engineer, in the June issue.

♦

Charles A. Powel, manager of headquarters engineering of Westinghouse Electric & Mfg. Co., has been elected president of the American Institute of Electrical Engineers.

♦

Erling Frisch, designer of propulsion equipment for the Navy, was one of the recipients of the Westinghouse Order of Merit. Among others who received the award were: James DeKiep, engineer on naval work; Everett McCandless, test engineer for power station apparatus; Enoch H. Turnock Jr., engineer on welding electrode development; and Edward I. Reed, who developed a new method for grinding porcelain insulators.

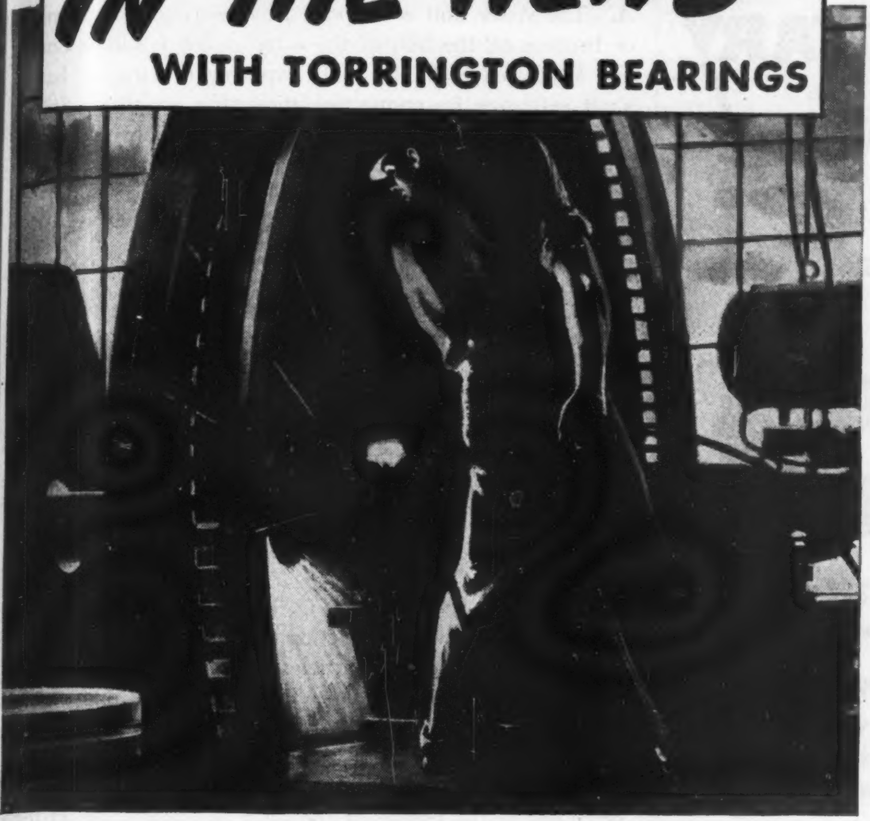
♦

Clinton E. Swift has been appointed assistant manager of the engineering and research department of Eutectic Welding Alloys Co. Before becoming connected with Eutectic, he had been manager of the welding division of Ampco Metal Inc.

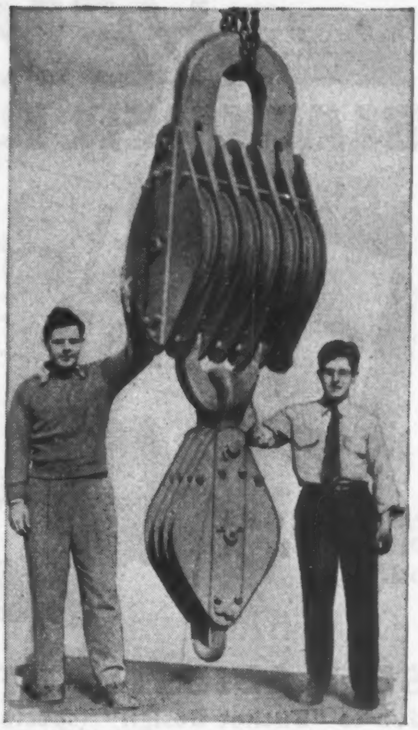
♦

Floyd T. Hague, formerly manager of engineering, has been made assistant to the vice president, acting as engineering consultant for all departments of the steam division of Westinghouse Electric & Mfg. Co., Philadelphia. C. B. Campbell was named manager of engineering, replacing Mr. Hague, and J. S. Newton was appointed assistant manager of engineering.

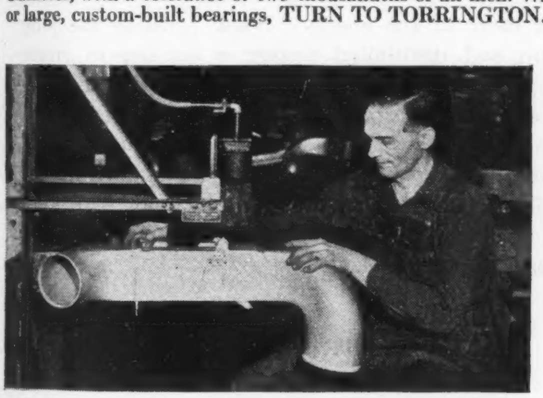
IN THE NEWS
WITH TORRINGTON BEARINGS

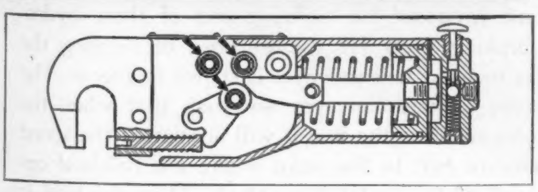


ALL IN THE DAY'S WORK typifies the attitude of Torrington engineers when they undertake the design and manufacture of anti-friction bearings for new or unusual applications. The skilled workman shown in the illustration is grinding the race for a precision bearing 10 feet in diameter, with a tolerance of two-thousandths of an inch. When you need counsel on standard or large, custom-built bearings, TURN TO TORRINGTON.

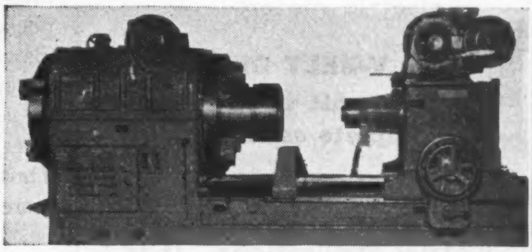


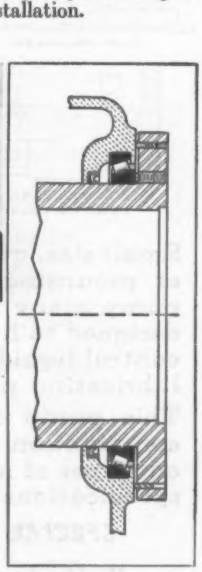
LIFTING 125-TON LOADS is the job of these tackle blocks designed and built by the Downs Crane & Hoist Company. With one block employing six sheaves, the other three, they are used in tandem to reeve the 19 parts of one inch wire rope required to sustain the load. Two NCS Needle Bearings, supplied by Torrington's Bantam Bearings Division, were installed in each of the sheaves which revolve on a 3″ hardened and ground shaft, with the pin hollow bored for pressure lubrication. NCS Needle Bearings combine the advantage of high load capacity and compact design with ease of installation.





THIS CINCINNATI PNEUMATIC Squeeze Type Riveter, with a 3-ton compressive force, manufactured by the Schauer Machine Company, provides an interesting application for Torrington LN Needle Bearings. Selected because of their compact design and high load capacity, the bearings were installed, as shown in the accompanying cross-section, at the points where pressure is extremely high.





MILLING THREADS in 8″ howitzers and 155 mm. guns is part of the important work performed by this Master Thread Miller, manufactured by the Smalley-General Company. To provide the essential accuracy at high speeds, and to take up the very heavy radial and thrust loads, both main and milling spindles are mounted in Tapered Roller Bearings, 30″ O.D., as shown in the accompanying cross-section. Eccentricity and face run-out of these bearings is .0005 maximum—an example of the ability of Torrington's Bantam Bearings Division to build precision bearings for heavy-duty applications.

ARMY E NAVY

TORRINGTON BEARINGS

STRAIGHT ROLLER • TAPERED ROLLER • NEEDLE • BALL

THE TORRINGTON COMPANY • BANTAM BEARINGS DIVISION

SOUTH BEND 21, INDIANA

Brakes

(*Continued from Page* 134)

A. The stator unit is supported by bearings, either roller or bronze, on the hub of the rotor which is either bored and keyseated to fit the shaft requiring braking, or has a shaft extension for connection through a coupling. Stator *B* is held from rotating. Suitable stuffing boxes prevent any leakage of fluid around the hubs.

How Fluid Friction Develops

On both sides of the rotor and stator are opposing pockets *E*, separated by the partitions *F* that are inclined to oppose the direction of rotation when braking is required. Cool water is introduced through the inlet pipe and the chambers *H* in the side walls and induced to flow into the pockets *E* by means of an arrangement of nozzles. During operation, the water which has been heated by fluid friction is discharged through the outlet pipe at the top and replaced by cool water from an appropriately placed supply tank.

Views *d*, *e*, *f*, and *g* of *Fig*. 8 show how this type of brake develops the fluid friction to which it owes its braking power. The white particles represent slugs of water. In view *d* the slug has been introduced into one of the rotor pockets and centrifugal force throws it to the outer circumference thereby imparting energy to it. In view *e* the slug of water is being directed with considerable velocity out of the pocket of the rotor and into those of the stator, but in so doing it is cut by the edges of the stator pocket partitions which are inclined to oppose the direction of rotation. This retards the speed of the rotor and dissipates some of the energy. View *f* shows the slug broken up and distributed among a number of stator pockets. In flowing back toward the hub, some of its remaining energy is dissipated by fluid friction against the stator walls. In view *g* the slug, still retaining some of its velocity, is being drawn back from the stator into the rotor. Here it is cut again, this time by rotor partitions. Thus, the velocity of the rotor is impeded further and more energy is dissipated.

Fluid Brake Applied to Conveyor

A scheme proposed for utilizing one of these hydrodynamic braking units for the purpose of limiting the speed of an inclined belt conveyor is shown in *Fig*. 9. The characteristics of this conveyor are such that when the motor is connected to the line it will accelerate the speed of the conveyor belt to the point where the full-load operating speed of the motor is reached. Then the load is increasingly applied to the belt at its upper end, accelerating it until the motor obtains synchronous speed. At this point an electric governor mounted on any one of the rotating shafts operates through a relay to disconnect the motor from the power lines, allowing it to idle and be driven by the conveyor load. The relay also acts to open a solenoid-actuated valve situated in the supply line of the braking unit. With the solenoid valve open, the pump is free to supply the braking unit with water from the

is one of many structural shapes being produced by American Magnesium Corporation. Strong because of its design and the alloy* used—very light in weight because it's magnesium—economical to employ because this extrusion replaces an expensive built-up beam.

There are doubtless many places where structural shapes having the combination of strength and light weight and economy will be advantageous. May we help you determine *where* in your products? Aluminum Company of America (Sales Agent for Mazlo Magnesium Products) 1703 Gulf Building, Pittsburgh 19, Penna.

***AM-C58S-T5**

MAZLO

MAGNESIUM PRODUCTS
REG. U. S. PAT. OFF.

AMERICAN MAGNESIUM CORPORATION

SUBSIDIARY OF ALUMINUM COMPANY OF AMERICA

supply tank through an adjustable plug valve. Inasmuch as heat is generated in the brake unit through fluid friction, a radiator and fan are required for the purpose of dissipating that heat.

Thus the complete water circuit is: Tank, through pump, through hydrodynamic brake unit, through radiator and back to tank. The amount of braking power exerted is dependent on the amount of water permitted through the braking unit and it will be seen that this amount can be controlled by the two adjustable plug valves. With the plug valves set so that the brake unit will hold the conveyor speed slightly above the synchronous speed of the motor, any further increase of conveyor speed due to overloading will be resisted by the brake unit.

Speed Control Is Automatic

Should the loading on the conveyor belt be decreased, the brake unit will slow the belt to a speed slightly below the synchronous speed of the motor at which time the governor will act to close the motor circuit and close the solenoid valve, allowing the brake unit to clear itself of water. Free of water, the brake unit becomes ineffective and the motor will continue to drive until its speed again rises to the cut-out point.

When braking units of this type are operated full of water (with maximum resistance) their capacities increase approximately as the cube of increases in speed. This explains their "governing" action in that a relatively slight increase in speed results in an appreciable improvement in power-absorption capacity. These units need not be operated full of water—their braking power at a given speed decreases in relation to the volume of water they contain. Thus it will be seen that any one size of such a braking unit will cover a considerable range of power capacity although, like any mechanism, its efficiency will be greatest under certain most favorable conditions.

Designer and Manufacturer Should Cooperate

In general, the brake types covered in this article are representative of those in most prevalent use. Special conditions of course require special solutions, but it is also true that the majority of braking problems are not unusual, and whole-hearted cooperation between the machine designer and the brake manufacturer invariably will result in satisfactory solutions. The designer is in a position to analyze the mechanical system and determine forces, timing, duty cycles, and kinetic energy and convey this information to the brake manufacturer so that the requirements may be compared with brake qualities. When no standard brake is suitable it usually is possible to design to meet the requirements, although obviously it is more economical to use commercial brakes wherever at all feasible.

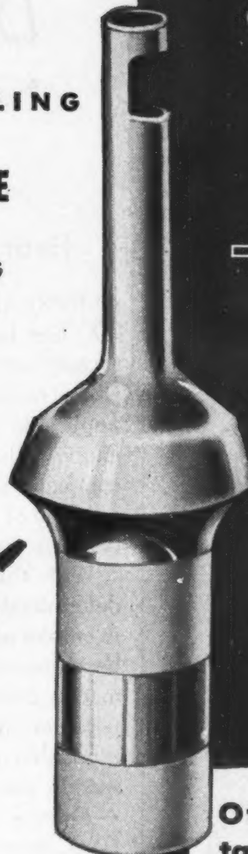
MACHINE DESIGN acknowledges with appreciation the collaboration of the following companies in the preparation of this article: Bendix Products Div., Bendix Aviation Corp. (*Fig.* 7); Cutler-Hammer Inc. (*Fig.* 1); The Electric Controller & Mfg. Co. (*Figs.* 2 and 3); Empire Electric Brake Co. (*Figs.* 5 and 6); General Electric Co. (*Fig.* 4); The Parkersburg Rig & Reel Co. (*Figs.* 8 and 9).

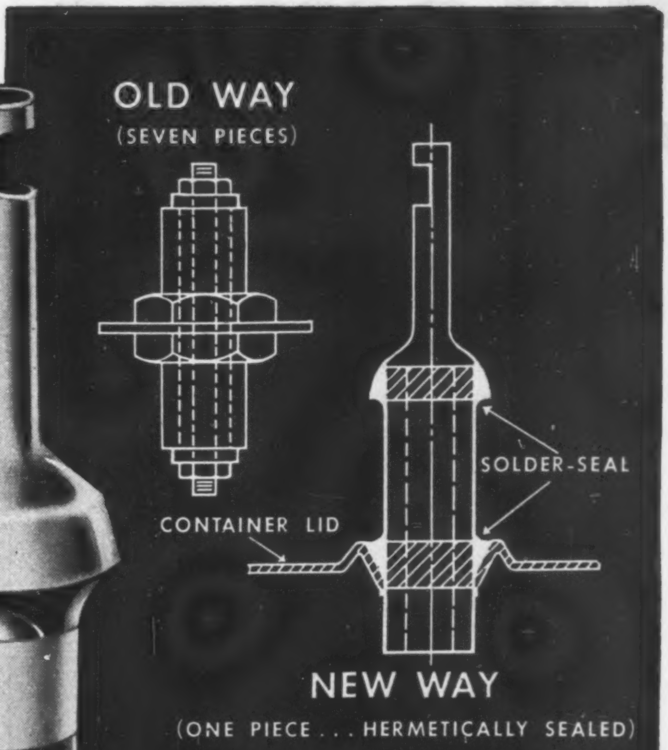
HERMETIC SOLDER-SEALING

MAKES PRESTITE

TERMINAL BUSHING

Leakage-proof





ACTUAL SIZE

High altitudes . . . humidity condensation . . . thermal shocks . . . cannot affect the performance of Solder-Sealed apparatus. The 100% hermetic bond assured by the metal-to-PRESTITE seal assures trouble-free service of terminal bushings.

The bushing consists of a PRESTITE tube on which are Solder-Sealed a terminal cap and a stud. Similar bushings are available without hardware for Solder-Sealing to other parts on the manufacturer's own production line.

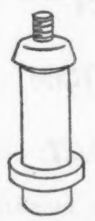
Solder-Sealed PRESTITE assemblies offer immediate help to manufacturers in many available standard forms. They also open up many new and added possibilities in postwar uses. For complete information, send for booklet B-3244. Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa., Dept. 7-N.

J-05142

Other PRESTITE methods of taking leads through partitions

APPARATUS ENCLOSING SOLDER-SEAL BUSHING—combination insulator, cover and terminal board—has a hollow construction which permits placing small devices inside.

SOLDER-SEAL ASSEMBLY—for vibrator packs, but can be used in similar apparatus, combining jack and terminal board.



SOLDER-SEALED BUSHING—for use with thicker gage covers of larger size transformers and capacitors. Bushing is Solder-Sealed to a metal ring which is soldered to the container cover.

PRESTITE is a dense nonporous ceramic compacted under high pressure and vacuum by the patented PRESTITE method of manufacture. This eliminates minute air pockets in the material, thus minimizing distortion in voltage gradients and eliminating internal corona discharges. PRESTITE is impervious to moisture and all chemicals except hydrofluoric acid. The quality of PRESTITE is consistently uniform, thus eliminating the need for the exaggerated safety factors common in other ceramics.

Westinghouse

PLANTS IN 25 CITIES . . . OFFICES EVERYWHERE

COMMUNICATIONS EQUIPMENT

INSTRUMENTS
D-C CAPACITORS
HIPERSIL CORES

DYNAMOTORS
RECTOX RECTIFIERS
INSULATING MATERIALS

DESIGN ABSTRACTS

Heat of Chips Measures Tool Force

SEVERAL research workers have recognized that the first law of thermodynamics—"When work is transformed into heat, or heat into work, the quantity of work is mechanically equivalent to the quantity of heat"—is applicable in metal-cutting operations. When an investigation of high-speed milling was initiated by the author's company, the calorimetric method was decided upon after a survey of available test equipment had been made. This decision was confirmed by a large number of tests.

With this method the power required by the tool is determined from the heat in the chips. Distilled water is employed as the medium for measuring the quantity of heat generated by a combination of friction and deformation during the cutting operation. By noting the temperature change of the water into which the chips fall, it is possible to study the effects of workpiece materials, feeds, speeds, and tool angles on power requirments of the tool. *—From a paper by A. O. Schmidt, Kearney & Trecker Corp., presented at the semiannual meeting of the A.S.M.E.*

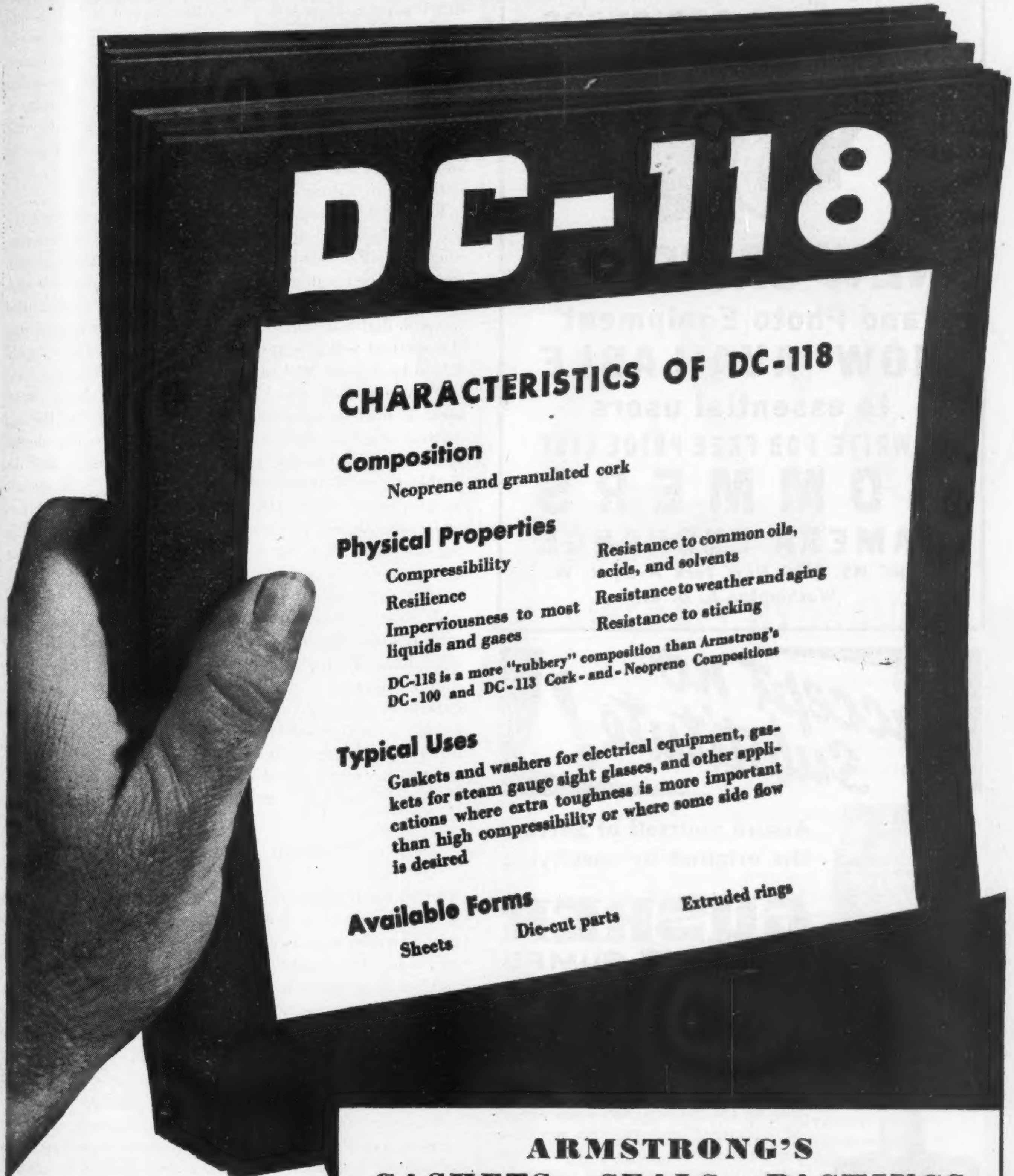
De-Icing Airplane Propellers

PREVENTION and removal of ice on airplane propellers have been attempted by mechanical, chemical, and thermal means. Experiments with elastic blade-covering devices, which were intended to stretch and thereby break the ice bond, have not given satisfactory results and do not appear practical.

Chemicals to be applied prior to flight have received extensive attention for a number of years. Recent results have justified the use in service of some compounds. The objections to this method seem to be the difficulty under service conditions in getting a proper application prior to flight, the loss in effectiveness due to time elapsed between application and use, and the effects upon performance of the airplane produced by applications of certain compounds. It can be expected, however, that preparations will be available which, if properly applied immediately before a flight, will provide an allowable minimum of protection for certain types and sizes of propellers during operations of limited duration.

Chemicals applied during flight have provided airline operators with an allowable minimum of protection on the propeller types and sizes which have thus far been employed in commercial aviation. Aids toward the efficient distribution of alcohol, which has been found by extensive service testing to be the most practical chemical, have been made available. These consist of a grooved-elastic covering, applied to the blade leading edge, which conducts the fluid to outer blade stations; and elastic connecting devices between the collector or slinger ring and

ARMSTRONG'S SEALING MATERIALS



DC-118 is one of more than fifty sealing materials developed by Armstrong. For descriptions of these materials, see Sweet's File for Product Designers or write us for a copy of the free booklet, "Gaskets, Packings, and Seals." Armstrong Cork Co., Industrial Division, 5109 Arch St., Lancaster, Pa.

ARMSTRONG'S
GASKETS · SEALS · PACKINGS

Cork Compositions • Cork-and-Synthetic Rubber Compositions
Synthetic Rubber Compounds • Cork and Rubber Compositions
Fiber Sheet Packings • Rag Felt Papers • Natural Cork

the blade leading-edge groove system. Effective operation of the alcohol system depends upon the alcohol being distributed prior to the icing encounter. The water droplets upon impact mix with the alcohol and, although some ice may form, the accretions are soft and easily thrown off without damage or annoyance to occupants.

Objections to the use of the alcohol system for propeller blades are that skill on the part of the pilot is necessary in order to get reasonably economical use of the fluid, that the system does not function satisfactorily in all types of ice, and that it is doubtful if satisfactory protection can be obtained on future larger-diameter propellers.

Electrical heating of the propeller blade through the use of a hub-type alternating-current generator and conducting rubber blade shoes has been investigated and found to possess practical possibilities. An accurate estimate of the practicability cannot be made from the limited research flights in icing conditions, but it does appear that this method will give protection at least equal to that obtained by the use of alcohol, and may weigh less and have less maintenance problems; however, this latter statement is open to question. Effectiveness of the thermal method is not as sensitive to icing types as is the alcohol system, and the skill of the pilot is not involved. It is desirable to avoid depending upon piloting skills to operate ice-prevention equipment because in icing conditions which are almost always accompanied by instrument flying conditions there are many other demands on the pilot.

Developments of other means of propeller blade heating for application to large diameter propellers and possibly rotating wing aircraft are under way. The present state of these developments does not permit a forecast of what form of application these will take. It does seem safe to forecast, however, that the protection for propellers of diameters greater than 15 feet will involve methods not as yet developed for propeller blades.—*From a paper by Lewis A. Rodert, National Advisory Committee for Aeronautics, presented at the National Aeronautic meeting of the S.A.E. in New York.*

Postwar or Long-Term?

POSTWAR planning is no new problem; it is no different, except in degree and in difficulty, from the long-term planning that successful business men are doing continuously. We should talk more about long-term planning and less about postwar. We are, it is true, facing a greater variety of new conditions and with less information which makes this planning just now more difficult; but, on the other hand, a highly favorable fact is that everyone realizes that conditions following the war will be different, and people will be mentally prepared for change.

Each kind of product, each customer group, and each market area has its past history, its war experience, and its postwar prospects. As always, these should be studied by familiar market-research methods. Separately, and in addition to this individual-product approach, each large company has its own history and experience through which it has developed an individuality and a composite character that can serve as an additional guide to its future prospects.—*From a paper by Frank D. Newbury, Westinghouse Electric & Mfg. Co., presented at the semiannual meeting of the A.S.M.E. in Pittsburgh.*

(*Concluded from Page* 154)

haps the engineers who find it necessary to figure volumes and weights so accurately have a working knowledge of

$$\frac{c}{2}=(2rh-h^2)^{1/2}$$

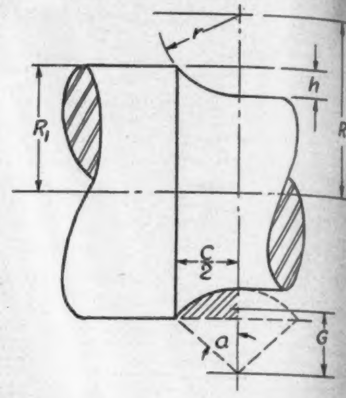
$$a=\sin^{-1}\frac{c}{2r}\,(degrees)$$

$$\frac{m}{2}=\frac{\pi ra}{180}$$

$$A=\frac{1}{4}[mr-c(r-h)]$$

$$G=\frac{c^3}{24A}$$

$$V=2\pi(R-G)A$$



where $c/2$=length of half chord; $m/2$=length of half arc; A=area of half segment (shown hatched); and V=volume of ring generated by the half circular segment.

Fig. 2—Formulas for calculating volume of filleted sections having arc less than 90 degrees

calculus, but no doubt there are some who would like to have the methods involving geometry and trigonometry.

—H. G. Taylor
Diamond Chain & Mfg. Co.

To the Editor:

May I express my appreciation to Mr. H. G. Taylor for his excellent letter in which he so clearly and ably offers constructive comments on my article? So far as I can see, his only criticism is that formulas derived from the simpler branches of mathematics such as geometry and trigonometry should have been presented rather than those derived by means of calculus.

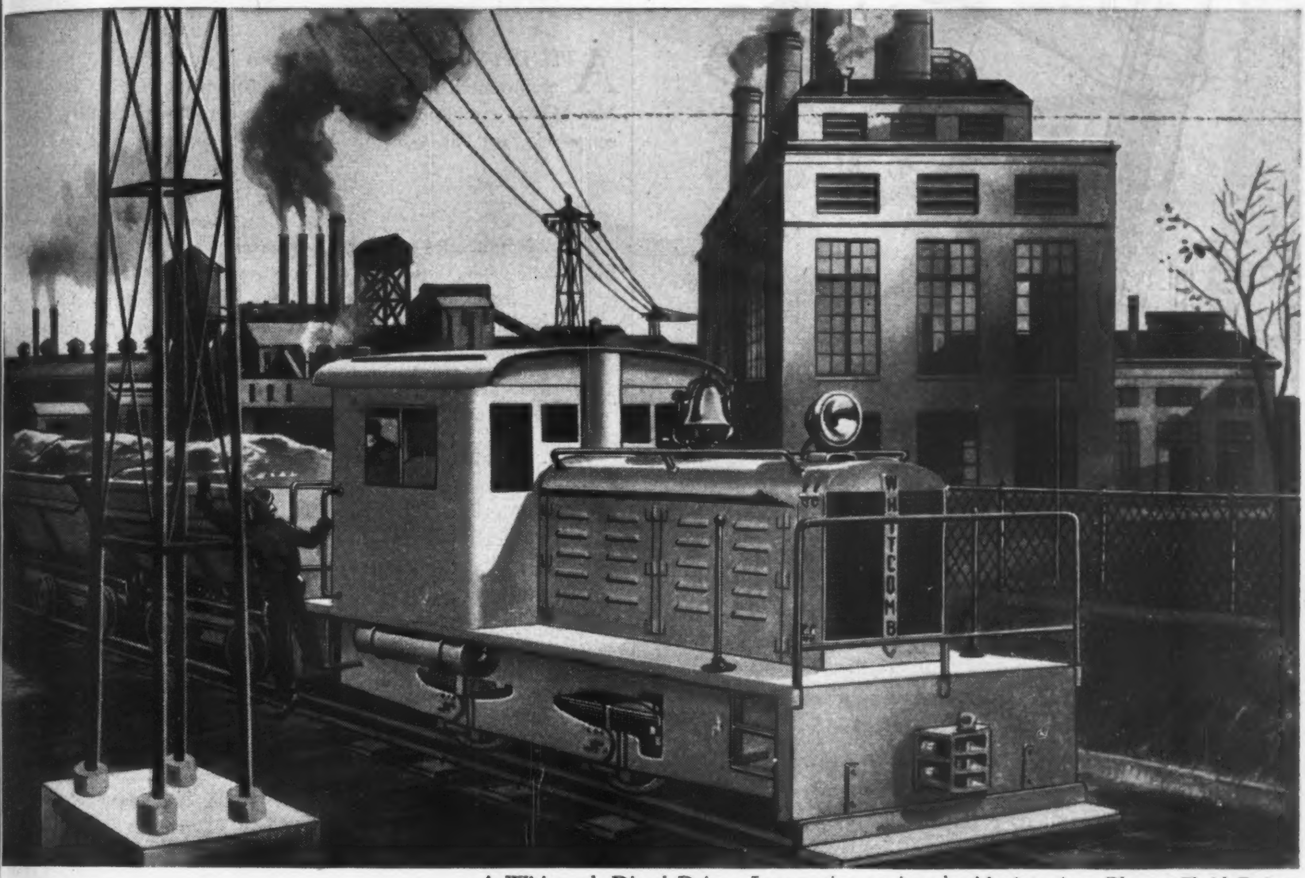
Being familiar with the methods to which Mr. Taylor refers, I gave them careful consideration when preparing the article. In the case of fillet sections having 90-degree arcs, the resulting equations are comparatively simple regardless of the method used. However, in the case of fillets with arcs less than 90 degrees the problem is, as Mr. Taylor admits, not so simple. When using the method he suggests it is necessary to calculate the exact area of a section through the fillet and the exact location of the "center of gravity" of this area before solving the problem. I felt that this method, although often used, is rather roundabout and that it would require as much work and almost as much mathematical knowledge on the part of the reader as the more logical and direct analytical method based on calculus.

Further, I felt that if the reader with limited mathematical training could not follow the calculus of the derivations, he would have little difficulty in handling the resulting equations which, except for their slightly unfamiliar form actually require nothing more than a working knowledge of elementary algebra and trigonometry for their solution.

—Clifford H. McClain
Llanerch, Penna.

WHAT!

FLUID DRIVE A LOCOMOTIVE?



A Whitcomb Diesel Driven Locomotive equipped with American Blower Fluid Drive.

Toughest job for any locomotive comes right at the start—moving the load.

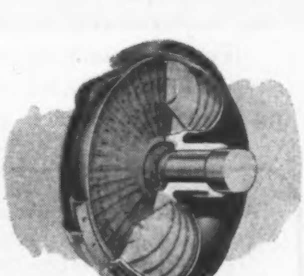
And that's where an American Blower Fluid Drive makes the big difference on a Diesel powered locomotive.

Fluid Driving through a hydraulic coupling on a Diesel locomotive materially increases the tractive effort and makes possible smooth acceleration from a standing start. It protects the Diesel Engine and transmission and improves over-all efficiency.

Right now all American Blower Fluid Drives are going to war as a vital part of fighter-plane equipment, in warships, cargo vessels, and into the plants of vital war industries.

After Victory investigate the advantages of Fluid Drive for your products or processes.

American Blower pioneered and developed the principle of Fluid Driving through hydraulic couplings in America.



Cutaway view of American Blower Fluid Drive. There is no mechanical connection between driving and driven members.

ARMY E NAVY

For excellence in production.

AMERICAN BLOWER

AMERICAN BLOWER CORPORATION, DETROIT, MICHIGAN
CANADIAN SIROCCO COMPANY, LTD., WINDSOR, ONTARIO
Division of American Radiator & Standard Sanitary corporation

BUSINESS AND SALES BRIEFS

APPOINTMENT of Wayne Martin as sales engineer has been announced by The National Smelting Co., Cleveland. Mr. Martin previously had been connected with Sperry Gyroscope Co. Inc. as assistant materials engineer.

•

Previously a sales specialist on welding products at American Brake Shoe Co., Dudley Rice has been made a field engineer in the Chicago district by Eutectic Welding Alloys Co. and will make his headquarters with two other field engineers at 53 West Jackson boulevard.

•

With headquarters in the Fisher building, Detroit, George F. Getschman has been named district sales manager for the Michigan, Indiana, Illinois, Wisconsin, Minnesota, Iowa, Missouri and northwest Ohio territory of Rolled Thread Die Co., Worcester, Mass.

•

Recent appointment of W. E. Cameron as national sales manager of Hydrotarder sales has been announced by The Parkersburg Rig & Reel Co., Parkersburg, W. Va.

•

Aluminum Co. of America, Pittsburgh, has appointed three assistant general sales managers—R. V. Davies will head sales engineering and development activities; R. B. McKee will be in charge of district sales offices and all direct selling activities, and Donovan Wilmot will be in charge of product manager activities and warehouse distribution.

•

Construction of a new building adjacent to the present plant at 1400 East 222nd street, Cleveland, has been started by the Mec-Rad division of Black Industries solely for the manufacture of mechanical and electrical components for radionics.

•

Formerly connected with Arrow-Hart and Hegeman Electric Co., Earl R. Sayre has been named application engineer for P. R. Mallory & Co. Inc., Indianapolis.

•

Change of name to Santay Corp. has been announced by Sinko Tool & Mfg. Co., Chicago, manufacturers of injection plastic moldings, sheet metal stampings, etc.

•

A new company to be known as Grayhill has been organized by Ralph M. Hill and Gordon E. Gray for the engineering and manufacture of mechanical and electrical switching devices used by electrical, electronic and aircraft concerns. General offices are at 1 North Pulaski road, Chicago, and

Business and Sales Briefs

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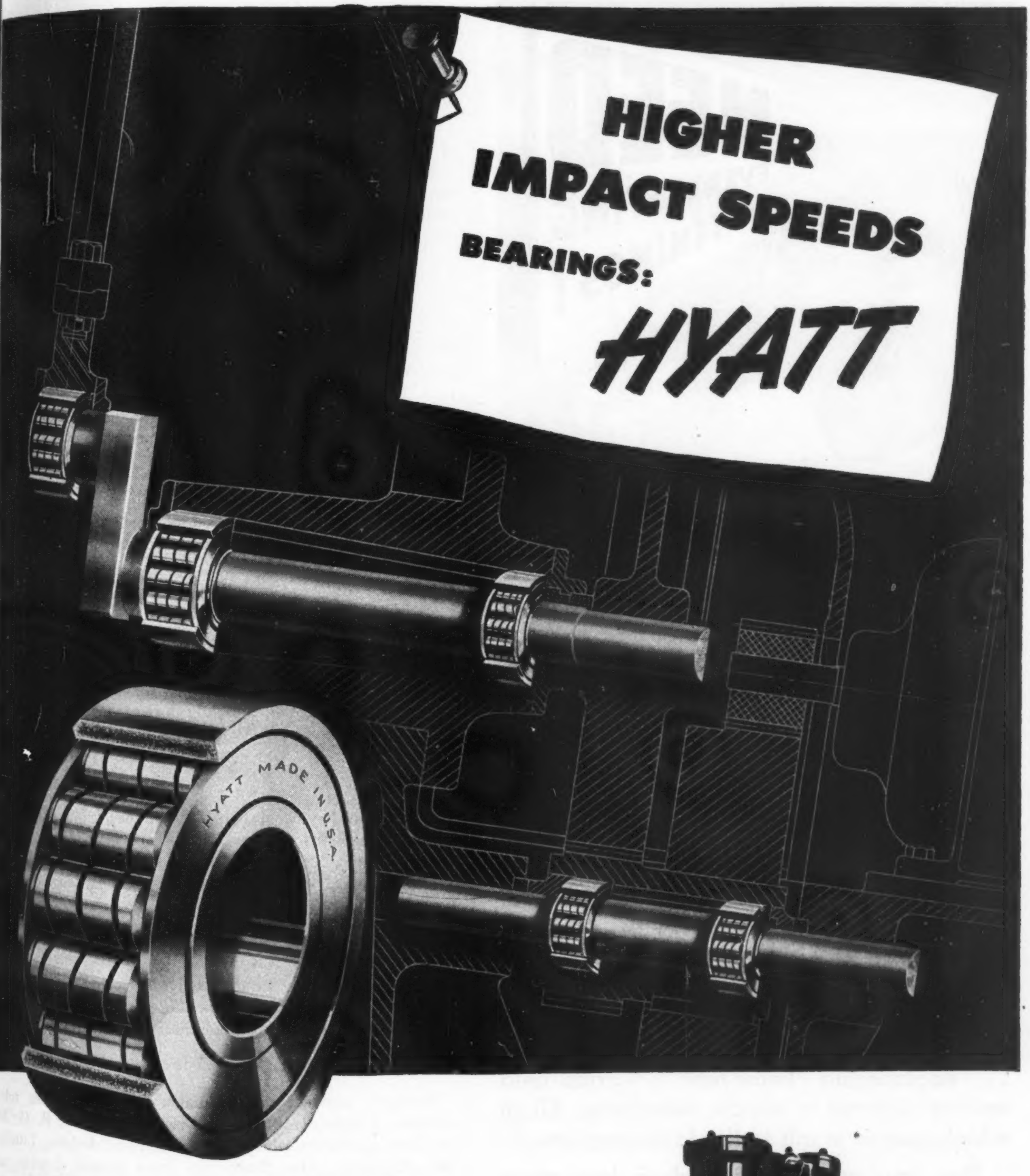


PNEUMATIC FORGING HAMMER

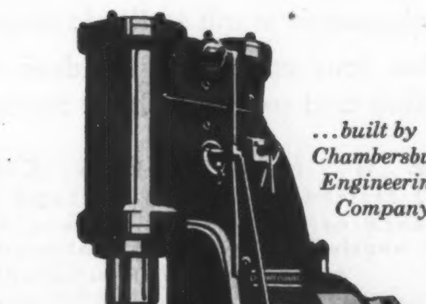
Crankshaft mounted on heavy duty Hyatt Roller Bearings, in separate case, permits assembly of complete drive unit.

Use of Hyatt Bearings eliminates need of bearing adjustments.

To economically forge present day tough carbon and alloy steels, more powerful blows, delivered with greater rapidity, are required of pneumatic forging hammers. The most dependable and efficient in heavy duty bearings is necessary to meet these severe conditions.

We will be glad to assist you with your bearings design problems.

HYATT BEARINGS *Division of* GENERAL MOTORS
HARRISON, NEW JERSEY



...built by Chambersburg Engineering Company

manufacturing facilities at La Grange, Ill. W. S. Lewis has been made chief mechanical engineer and general manager while Arnold Wassell has been placed in charge of plastic design engineering and production of plastic parts.

•

Succeeding Willard H. Cobb is Ernest G. Brown as general manager of mechanical goods, general products and Lastex yarn and rubber thread divisions of United States Rubber Co. Mr. Brown has been with the company since 1929.

•

At a recent meeting in Philadelphia the Pressed Metal Institute elected the following officers: F. C. Greenhill, vice president of Acklin Stamping Co., Toledo, O. as president; J. H. Robins, president of American Pulley Co., Philadelphia as first vice president; and Tom J. Smith Jr. of Cleveland and Huntington, W. Va., as executive vice president.

•

Purchase of an adjacent tract of land has almost doubled the present Newark, N. J., site of Mass & Waldstein Co., producer of industrial finishes.

•

Election of Charles O. Drayton as vice president in charge of sales and George H. Reama as vice president in charge of manufacturing has been announced by American Screw Co., Providence, R. I. Mr. Drayton formerly had been general sales manager while Mr. Reama had been factory manager.

•

T. R. Porter has been appointed technical-commercial representative on high-frequency heating by North American Philips Co. Inc. Mr. Porter, who had been associated with Westinghouse for the past eight years, will be located at company headquarters—100 East Forty-second street, New York.

•

Opening of offices in the Daily News building, 220 East Forty-second street, New York 17, has been announced by Van Doren, Nowland & Schladermundt, industrial designers.

•

Formerly national sales manager of the cellulose tubing division of Sylvania Industrial Corp., New York, R. G. Akin has been appointed sales manager, midwest division, Littelfuse Inc., Chicago. Dee Breen has been named division sales manager for the Western states and will make his headquarters at the plant in El Monte, Calif.

•

Appointment of Harry W. Gray as instrument sales engineer has been announced by Manning, Maxwell & Moore Inc., Bridgeport 2, Conn. Previously Mr. Gray had been assistant to the general sales manager, Airtemp division, Chrysler Corp., Dayton, O.

•

According to an announcement made by Shenango-Penn Mold Co., Dover, O., Paul H. Mielke has been made sales and service representative for the Pittsburgh territory covering western Pennsylvania, eastern Ohio (including the Youngstown area), and western New York (including Schenectady, Roches-

ter, Syracuse and Buffalo). Mr. Mielke, who has had thirty years in the centrifugal casting industry, will make his headquarters in Pittsburgh.

•

Titan Metal Mfg. Co., Bellefonte, Pa., has announced the removal of a building from New Castle, Del., to serve as an addition to the plants in Central Pennsylvania. As part of the expansion program a new melting, extruding and cold-finishing brass mill will be built and equipped. Production of brass rods is expected to double.

•

A new plant located at 30 East Burbank boulevard, Burbank, Calif., has been built by Ampco Metal Inc., Milwaukee 4, in order to improve its service.

•

Election of Carl I. Collins as executive vice president and director has been announced by Superior Steel Corp., Pittsburgh. Previously Mr. Collins had been executive vice president of Wickwire Spencer Steel Co. in New York.

•

According to a recent announcement, Alan P. Cline at 116 New Montgomery street, San Francisco 5, and Fred H. Williams at 5 Ivy street, Atlanta 3, Ga., have been appointed representatives in their respective territories by Avery Adhesives, Los Angeles.

•

Several recent additions to the sales force of The National Smelting Co., Cleveland, have been made recently. In production and sales work since 1936, V. D. Sweeney has been named sales manager. A. C. Hamilton has become Michigan representative while James F. Donnelly Jr. has been appointed eastern representative.

•

Lindsay & Lindsay, Chicago, maker of prefabricated metal structures, has appointed The Fielding-Wales Co. at 1836 Euclid avenue, Cleveland, as industrial representatives.

•

Purchase of an additional plant located at Fifth and Cutter streets, Cincinnati, has been announced by Newman Brothers. Plant No. 1 is at 660-670 West Fourth street, Cincinnati. The company at the present time is manufacturing valves and fittings, deck drains and sounding rods for airplane carriers, operating gear mechanisms, rudder bushings, hose connection flanges, stuffing boxes, etc.

•

According to a recent announcement by Goodyear Tire & Rubber Co., William L. Rodgers has joined the Mechanical Goods department and will handle V-belt sales in the Chicago territory. Mr. Rodgers previously had been employed by Dayton Rubber Co., Chicago, as an agricultural representative on V-belt sales.

•

Promotion of F. T. Turner to assistant sales manager of the brush division has been announced by The Osborn Mfg. Co., Cleveland. Sales and service engineer in the Pittsburgh area for several years and most recently in the home office

(*Continued on Page 194*)

Someone has to KNOW

COMPARATIVE BONDING QUALITIES OF RUBBER AND VARIOUS SYNTHETIC RUBBERS

Materials	Natural Rubber	Reclaim Rubber	Buna "S"	Buna "N"	Neoprene	Thiokol
Aluminum 2S and 3S	Good	Good	Fair	Good	Good	Poor
Black Iron	Good	Good	Good	Good	Good	........*
Brass 70/30 or less	Good	Good	Good	Good	Good	Fair
Bronze	Poor	Poor	Poor	Poor	Poor	Poor
Cadmium Plate	Good	Good	Good	Good	Good	........*
Cast Iron	Good	Good	Good	Good	Good	Fair
Chromium Plate	Fair	Fair	Fair	Fair	Fair	Fair
Copper	Poor	Poor	Poor	Poor	Poor	Poor
Dow Metal	Good	Good	Good	Good	Good	Good
Galvanized Iron	Poor	Poor	Poor	Fair	Good	........*
Lead	Good	Good	Good	Good	Good	Good
Stainless Steel	Fair	Fair	Fair	Fair	Fair	Fair
Steel	Good	Good	Good	Good	Good	Fair
Carbon	Good	Good	Good	Good	Good	........*
Wood	Good	Good	Good	Good	Good	........*
Celotex	Good	Good	Good	Good	Good	Good

NOTE: ORCO engineers are conducting additional tests on these materials. Complete information on all materials listed and many others is obtainable upon specific inquiry.

In planning the design, construction and operating efficiencies of machinery of all descriptions, where rubber and synthetic rubber must be bonded to metals and other materials, *someone has to KNOW* the adhesion qualities of the specified materials.

The above chart summarizes the results of exhaustive tests conducted by ORCO engineers on the bonding qualities of the indicated materials.

Limitations of space make it impossible to list the multitude of materials on which factual information on adhesion qualities is available from ORCO engineers.

This storehouse of accumulated experience is available to YOU. It is part of what we term "ORCO-OPERATION".

"ORCO-OPERATION"

THE OHIO RUBBER COMPANY · WILLOUGHBY, OHIO

BRANCHES: DETROIT · NEW YORK · CHICAGO · INDIANAPOLIS · WASHINGTON · CLEVELAND

(*Continued from Page* 188)

special assignments, Mr. Turner will organize sales planning and assist sales manager, M. C. Pecsok, in supervising the sales force.

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Election of the following presidents of subsidiary companies has been announced by Union Carbide & Carbon Corp., New York: Dr. Joseph G. Davidson, Carbide & Carbon Chemicals Corp.; James W. McLaughlin, Bakelite Corp.; Stanley B. Kirk, The Linde Air Products Co. and The Prest-O-Lite Co. Inc.; Arthur V. Wilker, National Carbon Co. Inc.; John D. Swain, Electro Metallurgical Sales Corp.; John R. Van Fleet, United States Vanadium Corp.

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H. L. Harvill Mfg. Co. has appointed Philip G. Nase and Henry J. DeBry as directors. Mr. Nase acts as eastern representative for the company and Mr. DeBry is secretary as well as office manager.

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With resident engineer Arthur D. Campbell in charge, a new district office in Cleveland has been opened at 623 Penton building by Vinco Corp., Detroit.

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According to a recent announcement, American Brake Shoe company has appointed Fred P. Biggs as vice president in charge of sales of the Brake Shoe and Castings division and the Southern Wheel division. Associated with the organization since 1916, Mr. Biggs will make his headquarters in New York. Stephen S. Conway, a member of the sales department since 1929, has been named assistant vice president of the same divisions and will be located in Chicago.

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Reelection of Robert L. Coe as president took place at the twenty-fourth annual meeting of the Copper & Brass Research association. Mr. Coe is vice president of Chase Brass & Copper Co. Inc., Waterbury, Conn.

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Goodyear Tire & Rubber Co. has announced that construction of a second addition to the fabric mill at Decatur, Ala., has been started. The first addition, which has just been completed, as well as the new one will be devoted to production of rayon fabric for tires and other rubber products.

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Succeeding I. K. Farley is V. H. Hiermeier as industrial manager of the St. Louis office of Brown Instrument Co., Philadelphia precision industrial instrument division of Minneapolis-Honeywell Regulator Co. Mr. Hiermeier has been connected with the instrument field for 15 years.

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Under the direction of Walter A. Fairchild, a new sales office has been opened by Bliss & Laughlin Inc. at 650 Main street, Hartford 3, Conn.

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Election of J. K. B. Hare as one of four new vice presidents has been announced by Westinghouse Electric & Mfg. Co. Formerly he had been manager of Central District sales activities with headquarters in Pittsburgh.

Designing Molded Plastics Parts: HEAT RESISTANCE OF PLASTICS

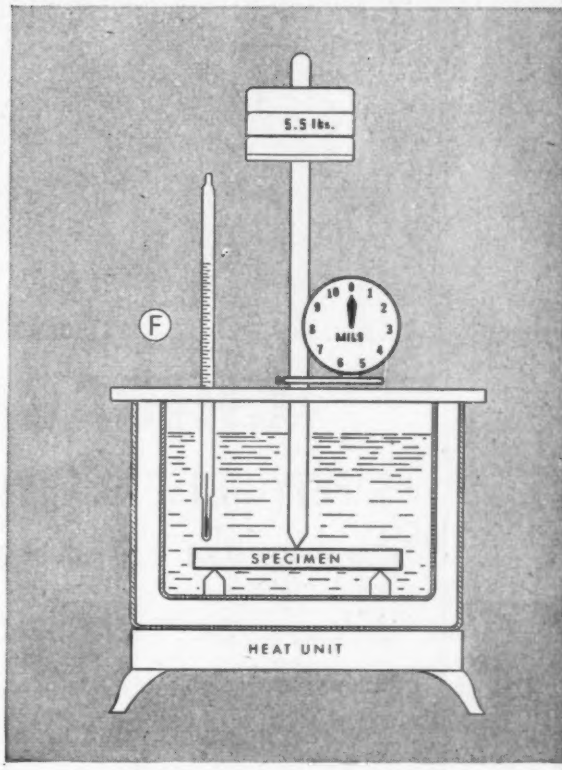
From the engineering files of One Plastics Avenue

The resistance of plastics to heat is a property which is hard to define. In order to work to a definite standard, the ASTM test D648-41T is used to define the **heat distortion point.** It is the temperature at which a 5 x ½ x ½ bar when loaded as a simple beam with a 5.5 lb. weight will deflect 10 mils. Since an actual application will seldom duplicate these conditions, this test is valuable only for comparisons between materials.

The **maximum safe continuous operating temperature** is the highest temperature at which the particular application or part can be used. A part is considered to be above its maximum safe continuous operating temperature if it tends to lose over 10% of its mechanical strength, or loses its surface appearance and becomes blistered, discolored, or distorted.

If a thermosetting material is subjected to continuous heat, it will not soften but will continue to function until it reaches its maximum safe operating temperature. If a thermoplastic material is heated, it tends to soften. The **softening point** is frequently evaluated as that point at which the plastic becomes readily deformed—as for drawing or shaping of thermoplastic sheets. It is usually above the maximum safe operating temperature.

When subjected to high temperatures, most plastic materials will ignite and burn. The **ignition point** is considered to be that temperature at which the material will ignite after 10 seconds exposure. This is the ultimate temperature which the material will stand. A plastic part could, if the exposures to high temperature were short and infrequent enough, operate safely up to the ignition point.



TEST FOR HEAT DISTORTION POINT

Material	Heat Distortion Point 0° F	Maximum Safe Continuous Operating Temperature for Molded Parts (under favorable conditions)	Softening Point	Ignition Temperature
Thermoplastics No load	130-190	140-190	150-260	600-900
Under load		120-170		
Phenolic Woodflour filled	240-250	240-250	None	1000
Asbestos filled	290-300	290-300	None	1150
Urea	260-270	170	None	1300
Mycalex		600	1100	
Cold Molded Non-refractory		500	None	
Refractory		1300	None	

REPRINTS of this advertisement may be obtained by writing Section E-220, One Plastics Avenue, Pittsfield, Mass.

Hear the General Electric radio programs: "The G-E All-girl Orchestra" Sunday 10 P.M. EWT, NBC. "The World Today" news, every weekday 6:45 P.M. EWT, CBS.

BUY WAR BONDS

PD-220

GENERAL ELECTRIC

MEETINGS AND EXPOSITIONS

Sept. 13-15—
Society of Automotive Engineers Inc. National tractor meeting to be held at Schroeder hotel, Milwaukee. John A. C. Warner, 29 W. Thirty-ninth street, New York, is secretary and general manager.

Oct. 2-4—
American Society of Mechanical Engineers. Cancellation of fall meeting scheduled to be held in Cincinnati has been announced. Clarence E. Davies, 29 West Thirty-ninth street, New York, is secretary.

Oct. 5-7—
Society of Automotive Engineers Inc. National aeronautic meeting and engineering display to be held at The Biltmore hotel, Los Angeles. John A. C. Warner, 29 West Thirty-ninth street, New York, is secretary and general manager.

Oct. 12-14—
American Society of Tool Engineers. Semiannual meeting to be held at Hotel Syracuse, Syracuse, N. Y. Adrian L. Potter, 2567 West Grand boulevard, Detroit, is executive secretary.

Oct. 13-14—
Electrochemical Society Inc. Fall meeting to be held at Hotel Statler, Buffalo. Colin G. Fink, 3000 Broadway, New York, is secretary.

Oct. 16-18—
American Institute of Mining and Metallurgical Engineers. Fall meeting of the Iron and Steel and Institute of Metals divisions to be held in conjunction with the National Metal congress, Public Hall, Cleveland. Convention headquarters will be at Hotel Statler. Frank T. Sisco, 29 West Thirty-ninth street, New York, is division secretary.

Oct. 16-19—
American Welding society. Annual meeting to be held in conjunction with the National Metal congress, Public Hall, Cleveland. Convention headquarters will be at Hotel Cleveland. M. M. Kelly, 33 West Thirty-ninth street, New York, is secretary.

Oct. 16-20—
American Society for Metals. National Metal congress and war conference displays to be held at Public Hall, Cleveland. Headquarters will be at the Statler and Hollenden hotels. W. H. Eisenman, 7301 Euclid avenue, Cleveland, is secretary.

Oct. 17-20—
Society for Experimental Stress Analysis. Semiannual meeting to be held in conjunction with the National Metal congress, Public Hall, Cleveland. Convention headquarters will be at Carter hotel. Additional information may be obtained from headquarters of the Society at Massachusetts Institute of Technology, Cambridge, Mass.

Oct. 18-20—
American Industrial Radium and X-Ray society. Annual meeting to be held in conjunction with the National Metal congress, Public Hall, Cleveland. Convention headquarters will be at Hotel Hollenden. Philip D. Johnson, 25 East Washington street, Chicago, is secretary.

Oct. 19-21—
Electronic Parts and Equipment Industry conference sponsored by Association of Electronic Parts and Equipment Manufacturers, Sales Managers club, Parts division of Radio Manufacturers association, and the National Electronics Distributors association, to be held at Stevens hotel, Chicago. Additional information may be obtained from Charles Golenpaul, publicity chairman, care of Aerovox Corp., New Bedford, Mass.

Nov. 13-14—
Society of the Plastics Industry. Annual fall convention and exhibit to be held at Waldorf Astoria hotel, New York. Additional information may be obtained from headquarters of the Society at 295 Madison avenue, New York.

Oct. 30-Nov. 1—
American Gear Manufacturers association. Twenty-seventh semiannual meeting to be held at Edgewater Beach hotel, Chicago. Newbold Goin, 301-2 Empire building, Pittsburgh, is manager-secretary.